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OPTIMAL ACTUATION RESEARCH AND STUDY

H. A. PRICE
D. L. GIESEKING

MISSLE AND SPACE SYSTEMS DIVISION
DOUGLAS AIRCRAFT COMPANY
SANTA MONICA, CALIFORNIA

TECHNICAL REPORT AFFDL-TR-67-46

JULY 1967

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AIR FORCE FLIGHT DYNAMICS LABORATORY
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AIR FORCE SYSTEMS COMMAND
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FOREWORD

This report was prepared by the Douglas Aircraft Company, Inc., Missile and Space Systems Division, Santa Monica, California, under Contract No. AF 33(615)-3514 and under the direction of the Air Force Flight Dynamics Laboratory, Research and Technology Division; Project No. 8225, Task No. 822510, and BPSN 6(638225 62405364). The Program Monitor is Vernon R. Schmitt/FDCL.

This report covers the period from 1 March 1966 to 1 January 1967, and is submitted May 1967 in fulfillment of the above contract. It is catalogued at Douglas as DAC-60576.

This technical report has been reviewed and is approved.

H. W. Basham, Chief
Control Elements Branch
Flight Control Division
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ABSTRACT

The servo actuating and corresponding energy distribution subsystem comprise approximately 80% of an aircraft's flight control system weight. Consequently, whenever subsystem tradeoff studies are conducted it is desirable to select the optimum design with respect to weight and other similar parameters. This study investigates and develops such an optimal design process. A sample problem was selected an an optimal technique formulated and applied to the problem. The sample problem was a fixed-configuration hydraulic actuation and power system. The study objectives were to optimize weight, dollar cost, size, dynamic performance, and reliability as a function of the system's independent design parameters. The parameters included pressure, actuator area, actuator torque arm, and plumbing tube sizes. Parameter optimization was accomplished by fixed grid and random searching techniques. Within the framework of parameter optimization, a design philosophy was formulated which allowed dissimilar terms (e.g., weight in pounds and dollar cost in dollars) to be combined to form a total performance criterion for the system. When the optimization technique was applied to the sample problem, the performance criterion showed little variation as a function of the parameters being optimized. All of the cost functions had large nominal values and only slight variations about this nominal. To realize the potential of the design technique developed in this study, different design concepts and possible variations of each one should be considered, to reach a more meaningful optimum design.

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ILLUSTRATIONS (Continued)

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SYMBOLS

PARAMETER	SYMBOL
Pump	PU
Weight	W
Flow	Q
Cost	C
Size	V
Reliability	R
Accumulator	ACC
Weight	W
Cost	C
Size	V
Reliability	R
Reservoir	RE
Weight	W
Cost	C
Length	L
Diameter	D
Reliability	R
Tubing	TU
Weight	W
Cost	C
Reliability	R
No. 1 steel	S1
No. 2 steel	S2
No. 1 aluminum	AL1
No. 2 aluminum	AL2
Total pressure loss at 120°F	DP
Total pressure loss at -40°F	DP4
Total fluid weight	FLW
Length	L
Flow	Q
Per foot of length	F
Inside diameter	ID
One-half maximum rate	Z
Maximum rate	M
Tubing diameter	D
No. 1 steel	S1
No. 2 steel	S2
No. 1 aluminum	AL1
No. 2 aluminum	AL2

SYMBOLS (Continued)

PARAMETER	SYMBOL
Tubing pressure loss (individual line)	DP
No. 1 steel at 120°F	S1
No. 2 steel at 120°F	S2
No. 1 aluminum at 120°F	AL1
No. 2 aluminum at 120°F	AL2
No. 1 steel at -40°F	S14
No. 2 steel at -40°F	S24
No. 1 aluminum at -40°F	AL14
No. 2 aluminum at -40°F	AL24
Tubing fluid weight (individual line)	TFW
No. 1 steel	S1
No. 2 steel	S2
No. 1 aluminum	AL1
No. 2 aluminum	AL2
Actuator	ACT
Weight	W
Length	L
Pressure drop (calculated)	DP
Pressure drop (assumed)	DP1
Stroke	S
Valve assembly	VAS
Weight	W
Weight at 3,000 psi	W3
Maximum pressure drop	DPM
Valve actuator package	VAP
Weight	W
Cost	C
Reliability	R
Fluid	FL
Temperature	T
Elasticity	E
Actuation system	
Mechanical linkage weight	MLW
Actuator cylinder elasticity	CYE
System elasticity	SE
Supply pressure (nominal)	PNOM
Supply pressure (off-nominal)	PRES
Valve flow per actuator	Q
Actuator piston area (nominal)	ANOM
Actuator piston area (off-nominal)	AREA

SYMBOLS (Continued)

PARAMETER	SYMBOL
Total hinge moment (per tandem actuator)	HMT
Hinge moment per actuator	HM
Design hinge moment per actuator	HMD
Torque arm length (nominal)	RNOM
Torque arm length (off-nominal)	R
Maximum control surface deflection	DELMX
Control surface velocity	DELD
Control surface moment of inertia	AYE
Resonant frequency	WN

SECTION I

INTRODUCTION

1. BACKGROUND

For each advanced aerospace vehicle, tradeoff studies are performed to develop the best design for the established requirements. So that the tradeoff studies are meaningful, the subsystem and system designer must have realistic information, particularly on weight and performance. In the area of servo-actuating subsystem design, this information often is based solely on an individual's educated guess rather than on an established technical method. This policy allows the bias from the designer's background and experience to influence the system design; also, often it is difficult to determine whether the selected design is optimum with respect to those factors that are used to judge a particular design, such as weight, performance, size, cost, and reliability.

The purpose of this study was to investigate the selection of parameters for an optimum, actuation-system design for a fixed-configuration system. This optimization procedure is commonly called "parameter optimization." This is only one small portion of the optimal design problem. For a truly optimal design, it would be necessary to consider different design concepts. Within the framework of parameter optimization, a design philosophy was formulated that allowed dissimilar terms (e. g., weight in pounds and dollar cost in dollars) to be combined to form a total performance criterion for the system. The results of this work were applied to a sample problem.

This study concentrates on the muscle portion of the servo-actuating subsystem; that is, the energy source and the valve actuator. The reason for deciding on this subsystem was that the majority of the flight-control system weight is associated with the actuation and power supply system. For example, the estimated weight breakdown for Douglas's proposed C-5A flight control system is as follows:

Pilot Controls	Automatic Flight Control System	Actuation Systems	Flight Control Hydraulic System
3%	16%	40%	41%

With regard to the titles in the above chart, PILOT CONTROLS and AUTOMATIC FLIGHT CONTROL SYSTEM are self-explanatory; however, ACTUATION SYSTEMS and FLIGHT CONTROL HYDRAULIC SYSTEM need additional explanation. Actuation Systems refer to all components that pertain directly to the aileron, elevator, and rudder actuation systems. Flight Control Hydraulic System refers to all components primarily in the hydraulic power supply system that do not pertain directly to any individual actuation system.

2. APPROACH

The following paragraphs describe the organization of the plan used to develop the optimal design process.

a. Determination of the Present State-of-the-Art

A survey was conducted to determine the status of actuation system design in the aerospace industry. The actuation systems were restricted to those having torque outputs and included hydraulic, pneumatic, and mechanical systems used in the industry.

b. Selection of a Sample Design Problem

Based on the data obtained from the survey, an actuation design problem was chosen for use in the development and application of the optimization technique.

c. Generation of a Mathematical Model

Because the optimal technique is an analytical process, it was necessary to generate mathematical relationships of the sample problem correlating its design parameters with the system outputs being optimized.

d. Formulation of a Specific Optimization Technique

The parameter optimization process best suited to this problem had to be selected and a method formulated for combining the various system outputs so that meaningful results could be obtained.

e. Integration of the Optimization Technique With the Design Problem

A theoretical system design utilizing the sample problem and the desired optimization technique was demonstrated with the aid of a computer. In this way, general design guidelines were established and solutions to major problem areas could be emphasized.

f. Evaluation of the Derived Design Technique

The usefulness of the new design technique was evaluated by comparison with current design methods and with any related efforts outside of Douglas. A review council, comprised of experienced Douglas design engineers, was established to provide counseling during the study and evaluation of the results upon completion of the study.

SECTION II

DESIGN PROBLEM

1. STATE-OF-ART SURVEY

The actuation system state of the art was established by a literature search. Its purpose was to determine the state of the art of servo actuation techniques in contemporary aerospace flight-control systems. The ground rules, made to regulate the extent to which the various aircraft, missile, and booster actuation systems were investigated, are as follows:

a. Only those systems that had torque outputs were considered.

b. The choice of an actuating system often depends on its required power supply; therefore, the power supplies also were included.

c. Only the valve actuator and load were considered in the servo actuation portion of the system. Amplifiers, feedback tranducers, and other electrical components fall logically into another category and were not included. The power distribution and usage portions were the elements included in the survey (Figure 1).

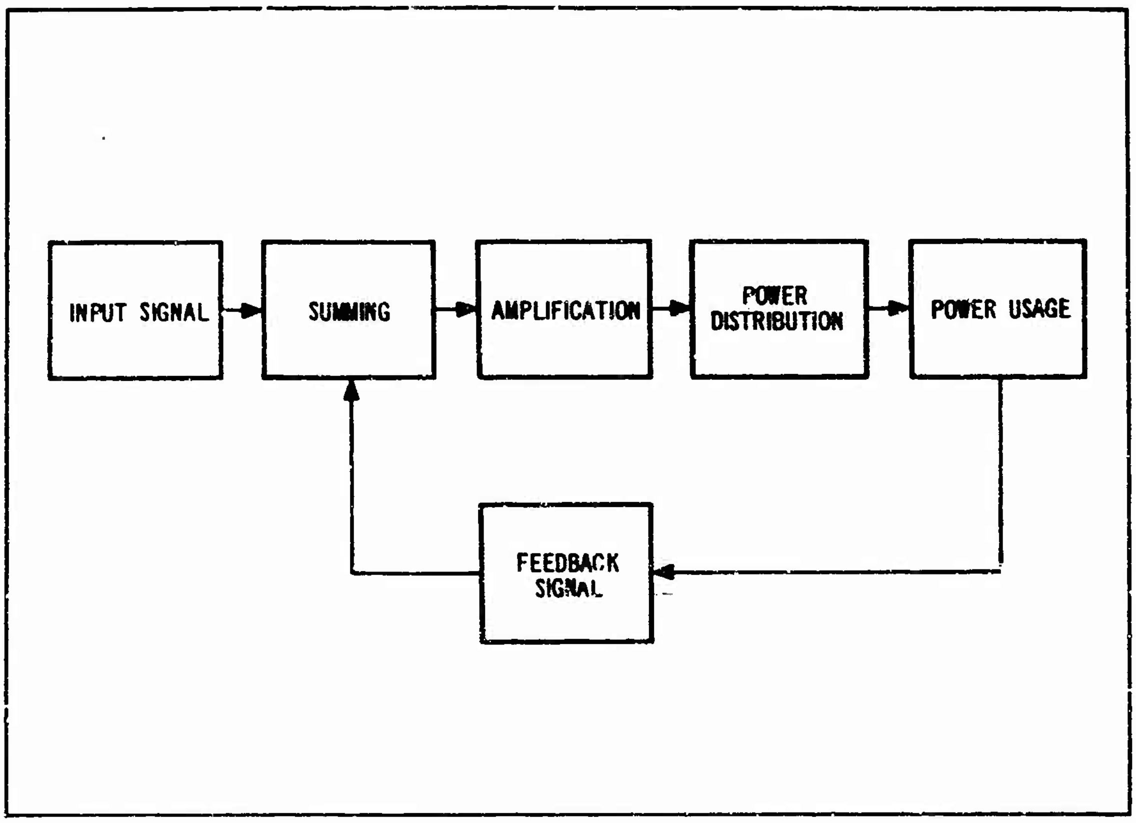

Figure 1. General Flight Control Actuation System

The type of systems investigated were divided into three basic design concepts: (1) electrohydraulic, (2) electropneumatic, and (3) electromechanical. Because of the amount of information collected, it is impossible to include all of it here. Essentially, the information consisted of a general comparison, a review of current and advanced technology, and a discussion of problem areas for each concept. With regard to hydraulic systems, advanced technology is primarily concerned with reliability improvement, efficiency improvement, and compatibility with high-temperature environments. Its principal problem areas are associated with temperature sensitivity, contamination sensitivity, storage, serviceability, and multiple-energy conversions. With regard to pneumatic systems, compressibility is the major problem area. The advanced technology is primarily concerned with the reduction of actuator volumes and the use of various types of feedback compensation. With mechanical systems, clutching is the major problem area, and as a result, these systems are limited to those having relatively small loads. Therefore, advanced technology is mainly concerned with the development of clutching arrangements capable of handling larger loads.

The results indicated that hydraulics is the most universal design concept and the most widely used type of actuation system in the aerospace industry. Hydraulics has the unique capability of combining the flexibility of pneumatics with the stiffness of a mechanical device, thus making it adaptable to most applications. The extensive use of hydraulic actuation systems and the large quantity of design information available on hydraulics led to the recommendation that a hydraulic concept be used for the optimization study design problem. Because of the greater familiarity with hydraulics in the field of actuation, fewer problems were anticipated in the application of new design techniques. Furthermore, a design study based upon the hydraulic concept will yield a more meaningful evaluation.

2. SAMPLE PROBLEM

The flight control systems of piloted aircraft are undergoing an evolution. Historically, the reliable primary flight control system consisted of a mechanical link between the pilot and the control surface. With the advent of supersonic and hypersonic aircraft, however, it is becoming more difficult to cope with the problems associated with a mechanical linkage system, such as backlash, friction, inertia, elasticity, and other nonlinearities. As a result, considerable effort is being expended in the development of alternate control schemes with adequate reliability, that is, fly-by-wire systems. In this type of control scheme, redundancy is used to obtain the reliability previously associated with a mechanical linkage system. An example, that emphasizes the problems associated with a mechanical linkage system is the flight-control system proposed by Douglas for the C-5A. This system includes a fly-by-wire mode of operation so that it can meet the terrain-following requirements. These requirements are so severe that it is not possible for the mechanical linkage system, by itself, to provide the necessary performance.

The newly designed subsystems and components for a fly-by-wire system will have many stringent requirements, such as high reliability, low cost, minimum weight, and dimensional limitations, and must be able to operate normally over a wide environmental range. These factors, plus the fact that the actuation and power supply systems dominate the flight-control system weight, resulted in the decision to use an aircraft system for the design problem. Also, because Douglas had recently proposed a flight-control system for the C-5A, it was decided to use the C-5A hydraulic-system design requirements as a guide in establishing the problem. Generation of an optimal design technique was considered more important than optimizing a particular system, so it was further decided to select only a portion of the flight-control actuation system and to simplify it to only the fundamental characteristics. After an optimal technique and corresponding procedural guidelines have been established, recommended future work will consider optimization of an actual system. In line with these decisions, a simplified version of the C-5A elevator actuation system was chosen for the design problem. A schematic of this simplified system is shown in Figure 2. Its essential features include the following:

a. A tandem valve actuator combination.

b. Dual control surfaces, thus requiring four separate valve actuators.

c. One hydraulic power supply consisting of a pump, reservoir, accumulator, and a considerable length of plumbing.

d. Mechanical linkage between the pilot and control surface as a backup for electrical input signals.

The vehicle performance requirements that apply to the elevator system and used in this study are the following:

a. A $+25^\circ$ to -15° maximum control-surface deflection.

b. A maximum control surface rate of 40° per sec.

c. A maximum hinge-moment capability of 48,000 in. lb per actuator per piston.

d. Design considerations at normal operating fluid temperatures include a maximum rate at zero hinge moment and one-half maximum rate at three-quarters maximum hinge moment.

e. A design consideration at -40°F fluid temperature of one-quarter maximum rate at zero hinge moment.

f. A bandwidth of 10 rad/sec and an overdamped step response with a 0.1 sec rise time.

In this design problem, the requirements will remain fixed and will constitute the minimum performance that the actuation system must meet, regardless of its design.

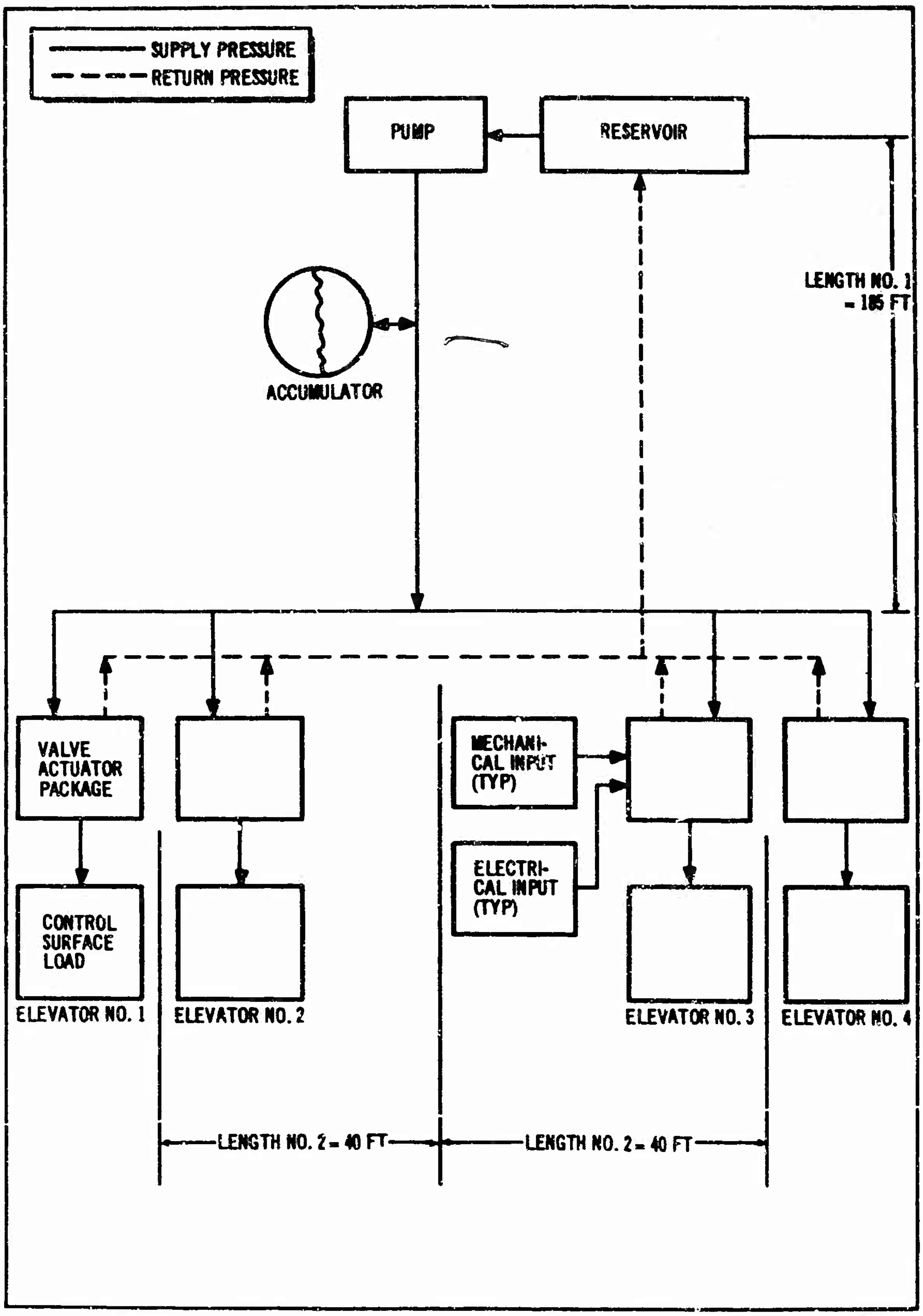

Figure 2. Hydraulic System Schematic

The actuation system dependent variables that will be optimized and that are called vehicle constraints are the following:

a. Weight.

b. Dollar cost.

c. Size.

d. Reliability.

e. Dynamic performance.

These constraints were selected arbitrarily. Other factors could be added to this list; however, these five seemed to be the most pertinent at the time the study originated.

The following actuation system design parameters that will be considered as independent variables in the optimization process are:

a. System pressure having a range of values from 2,000 psi to 6,000 psi.

b. Actuator piston area having a range of values from nominal (that area required to meet the performance requirements) to a +2 sq inch.

c. The length of the torque arm between the actuator and the load having a range of values from 4 in. to 8 in.

d. Fluid temperature having a range of values from -40°F to +200°F.

e. The plumbing tube sizes have a range from 0.5 to 1.0 in. OD.

3. MATHEMATICAL MODEL

For application of an analytical technique to the design of hardware, mathematical relationships must be formulated that relate the hardware design parameters to the end product. In this case the end product is a theoretical design having an optimum combination of weight, cost, size, reliability, and dynamic performance. For example, the weight of each component being considered will be related to the applicable system design parameters. A combination of all the necessary relationships produces the mathematical model. When combined, the relationships which are independent of time, are termed the steady-state model. Those which depend on time are called the dynamic model. The boundary conditions or assumptions used to guide the formulation of these mathematical models include:

a. The hydraulic system state of the art is based upon a nominal supply pressure of 3,000 psi. For pressure greater than 3,000 psi, a penalty factor is added so that approximately the same factor of safety is retained.

b. The same component configuration will be used, regardless of variation in the independent design parameters.

c. The effect of the pump power requirement upon the vehicle will not be considered.

d. The valve actuator supporting structure will be considered a rigid body.

e. The torque arm between the actuator and the load will be considered a rigid link with its length being the only item of interest.

f. The effect of development time, experience, quantity, expected life, and detail design will not be considered on any of the vehicle constraints.

g. A size factor will be used instead of volume because the volume of some components is not too meaningful. This characteristic has to be judged on the basis of the type of vehicle being used; for example, transport versus fighter vehicle.

h. The electrical portion of the flight-control system is not being considered.

i. The dynamic response of a hydraulic actuation system is not the same for different input amplitudes. As a result, there could possibly be an optimum dynamic performance associated with each input amplitude and so the decision was made at the beginning of the study to simplify this requirement and limit investigations to 10% inputs.

a. Steady-State Model

The steady-state model is separated into sections, each of which represents an individual component or subassembly. The required relationships are developed and the corresponding output equations for each section are summarized in Appendix I. Each of the various component and subassembly configurations will be described briefly. Obviously, more or less sophisticated relationships could have been used and different assumptions made. However, the fact that they have been considered and used is of prime importance because they provide the ground work upon which the optimization structure must be built.

(1) Pump

The pump is a variable-delivery, variable-angle unit with a rated speed of 3,600 rpm. A photograph of this pump is shown in Figure 3. Pump design is based on a working pressure of 3,000 psi. As a result, when supply pressures less than 3,000 psi are used, an over-designed condition will occur and a weight penalty realized. For pressures greater than 3,000 psi, an appropriate correction factor is added to the basic 3,000 psi relationship.

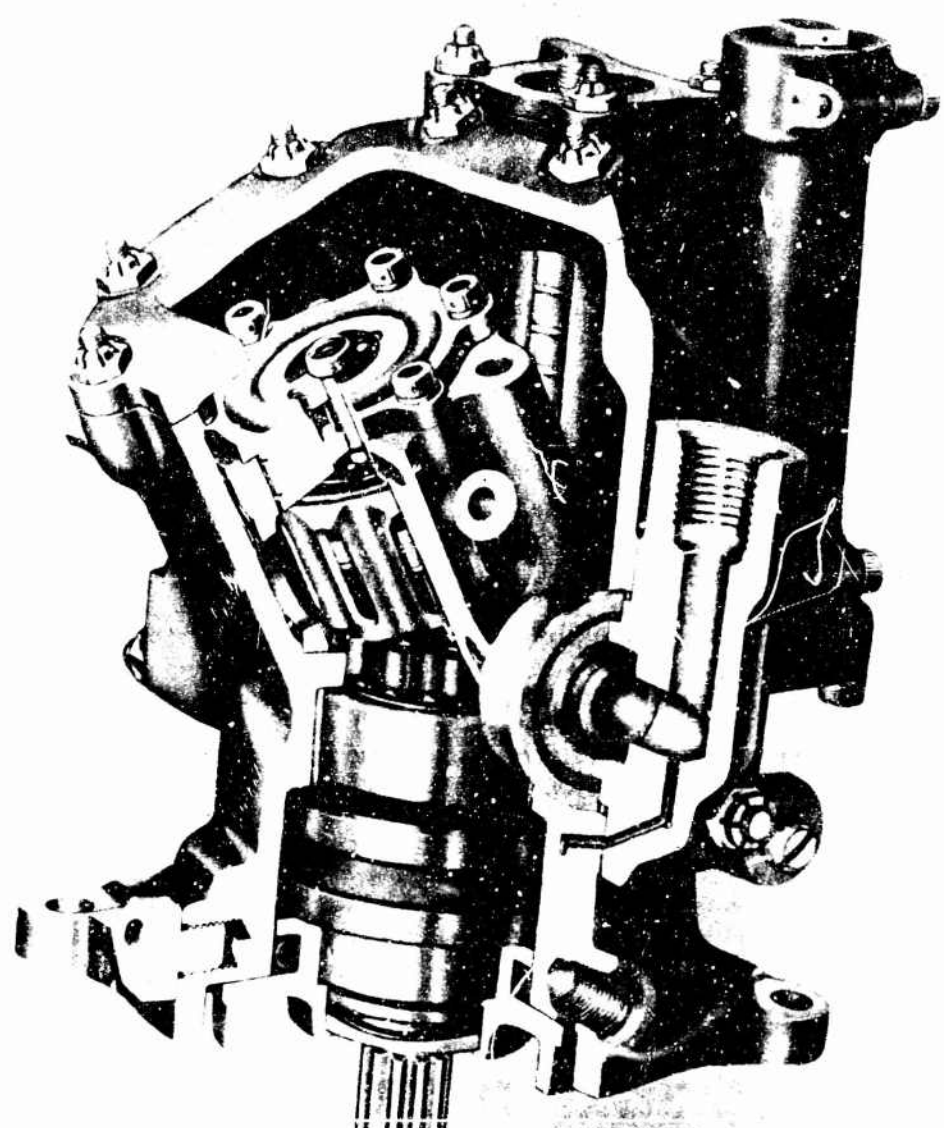

Figure 3. Variable Delivery, Variable Angle Pump

(2) Accumulator

Cylindrical piston type and spherical bladder type accumulators are available for this type of application. Because Douglas originally proposed the spherical type for the C-5A, it will be used in this study. Familiarity, successful utilization in past applications, and lighter weight are the primary factors which influenced the decision in favor of the spherical accumulator. A photograph of a typical unit is shown by Figure 4.

Because no limit exists upon the duration of the aircraft's requirement for maximum power, the pump is sized to accommodate the total required power at rated revolutions per minute and the accumulator will be used to minimize the effect of pressure disturbances in the hydraulic system.

(3) Reservoir

The reservoir will be a boot-strap type unit. It eliminates the adverse effects in the hydraulic system caused by changes of aircraft attitude and acceleration loadings, and because it is airless, does not contribute to the undesirable quantity of air dissolved in the hydraulic fluid. A photograph of this type of reservoir is shown by Figure 5.

(4) Tubing

The supply and return system tubing has been divided into two sections. Section No. 1 (as shown in Figure 2) is from the pump to the place where the plumbing branches out to each elevator valve actuator package. Section No. 2 includes all of the tubing from the branch point to Elevators 1 and 2 or 3 and 4. Tubing sizes can vary between sections and between the supply and return systems; however, within a section, the sizes are constant. Steel tubing is used for the supply system and aluminum for the return system.

(5) Valve Actuator Package

The valve actuator package is tailored after the unit used by Boeing for the 727 elevator-actuation system. Figures 6 and 7 are photographs of this unit. The various relationships that pertain to this unit are divided into the following groups: actuator, valve assembly, and valve-actuator package. Even though the valve and actuator are tandem units, the one hydraulic system will be used to supply both parts of these units. Also, for simplification, the same input signal will be used for each elevator so that the flow is equally divided.

(6) Mechanical Linkage

The C-5A flight control system included a mechanical link between the pilot and the elevator control surfaces. Because the weight of the mechanical link was about 15% of all the components and subassemblies directly associated with the elevator system, the linkage weight was included in the model. None of the independent parameters influenced its design; therefore, the weight was considered a constant. A schematic is shown by Figure 8.

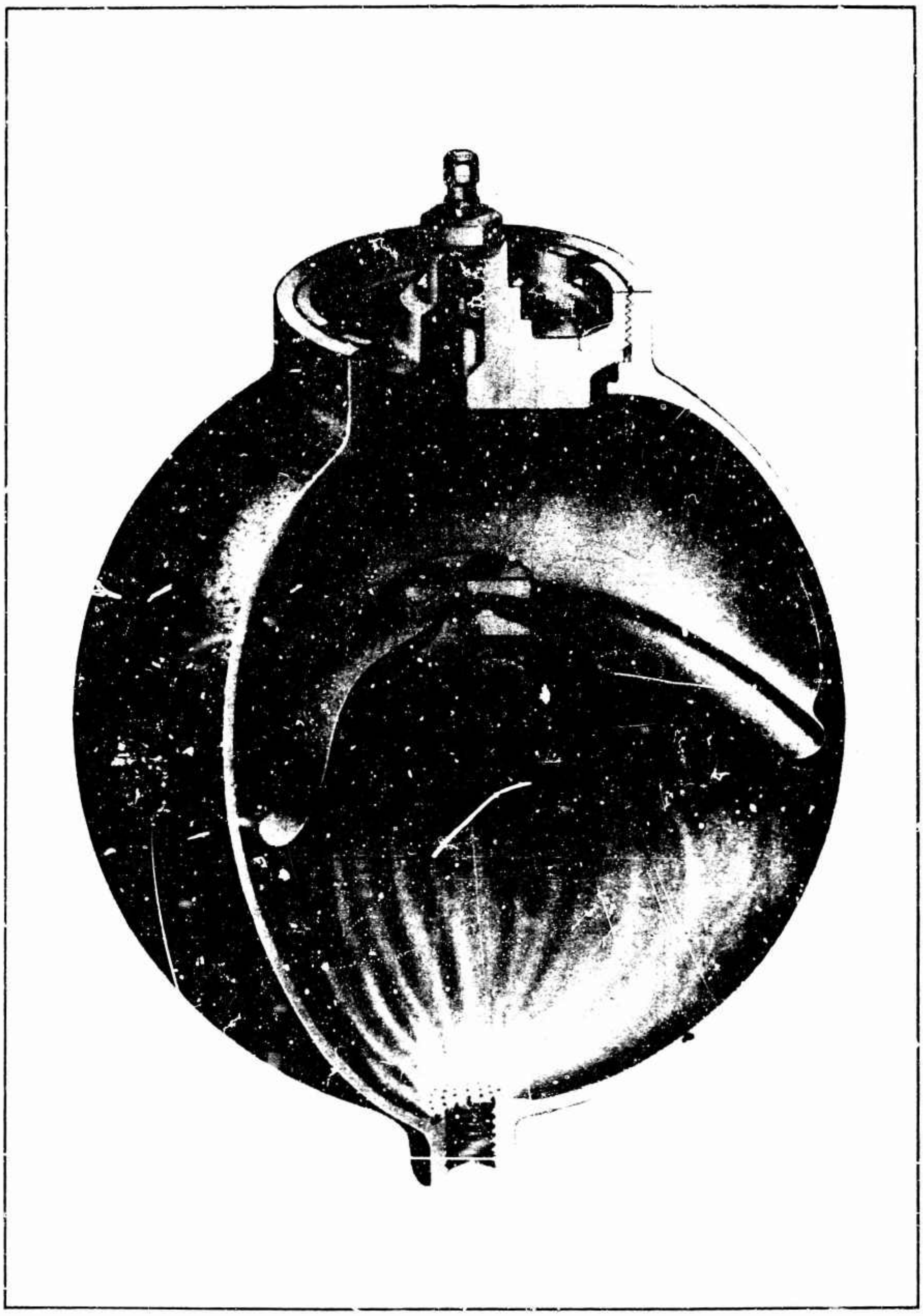

Figure 4. Spherical Type Accumulator

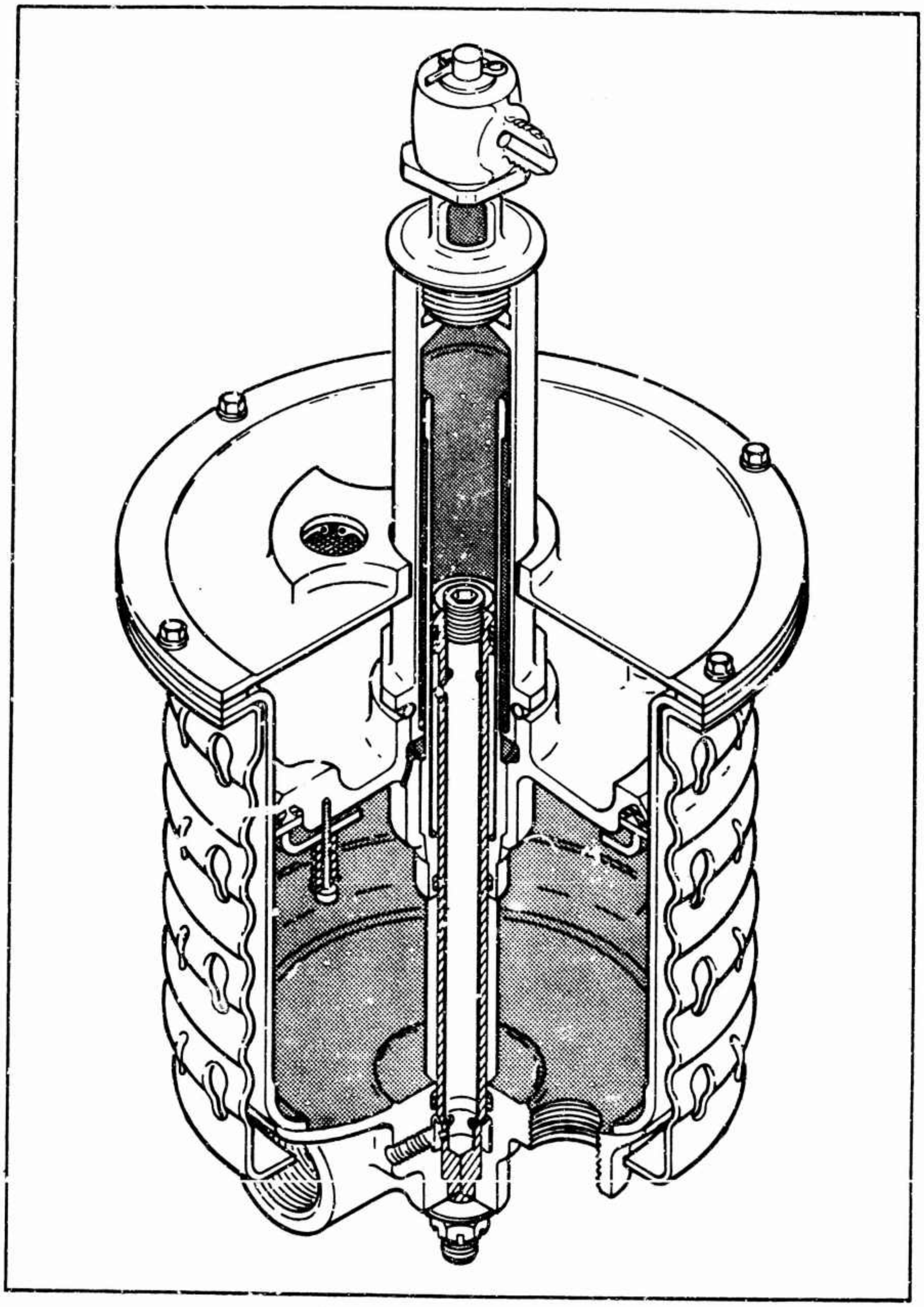

Figure 5. Boot Strap Reservoir

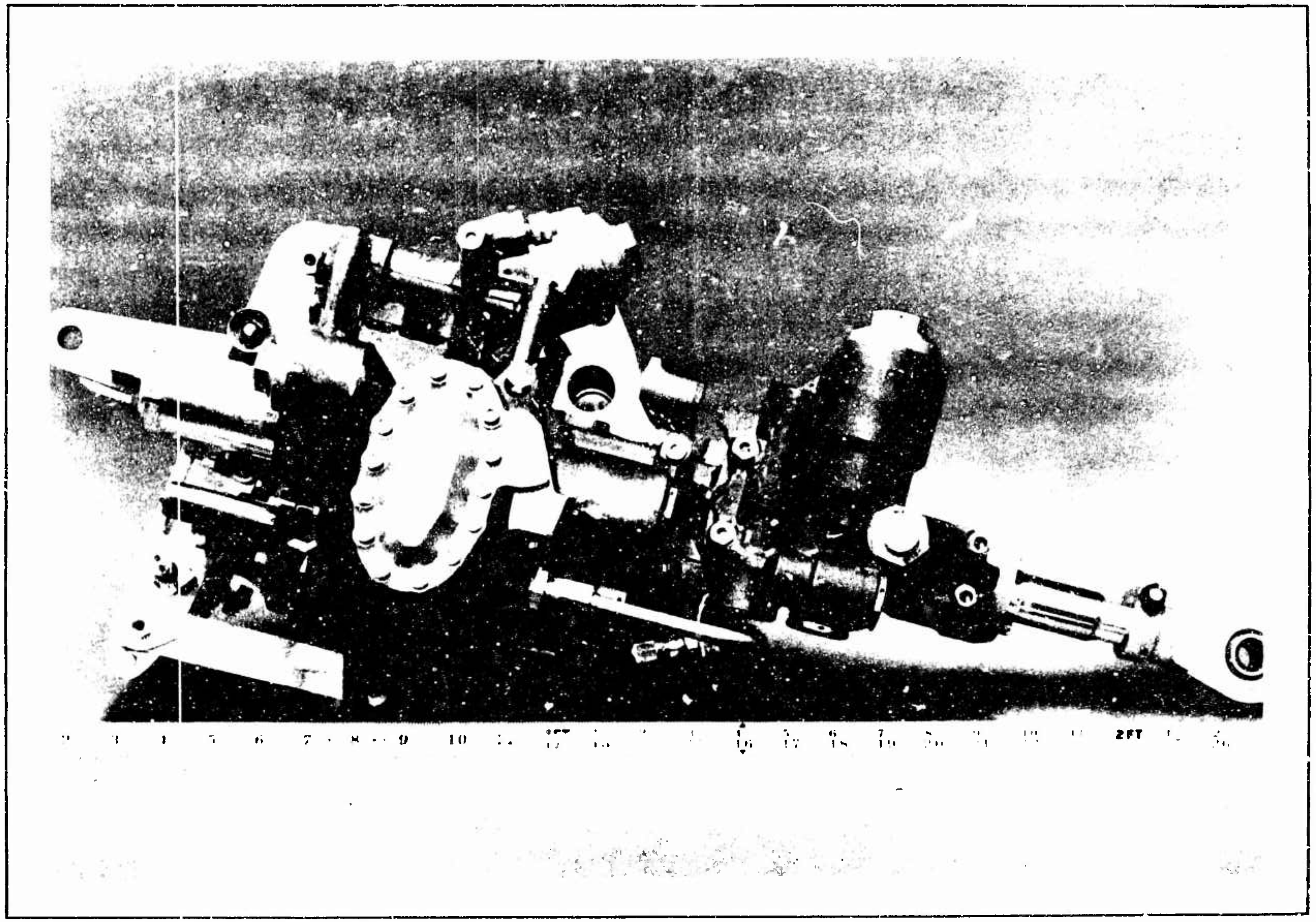

Figure 6. Valve Actuator Package – Side View

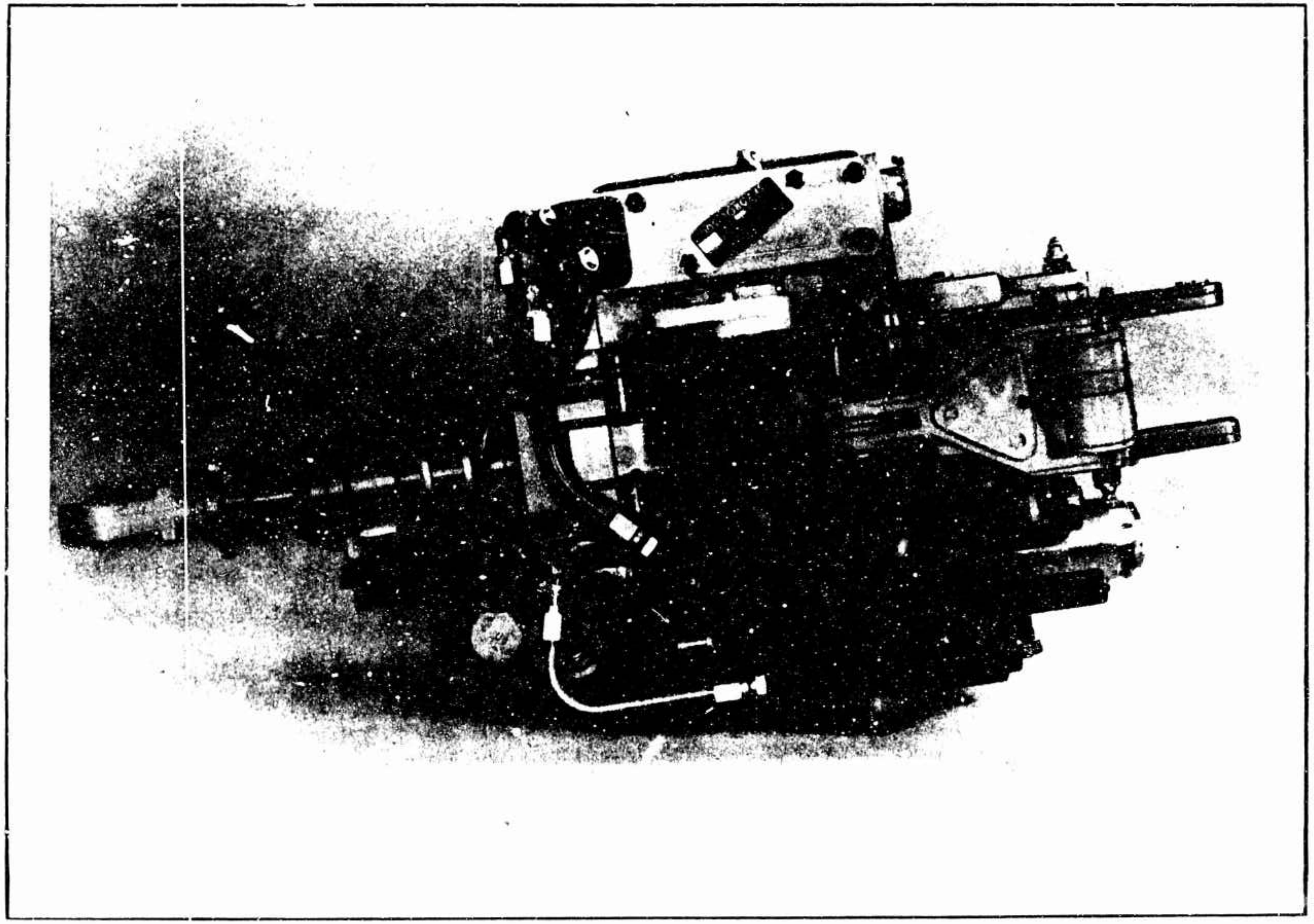

Figure 7. Valve Actuator Package – Too View

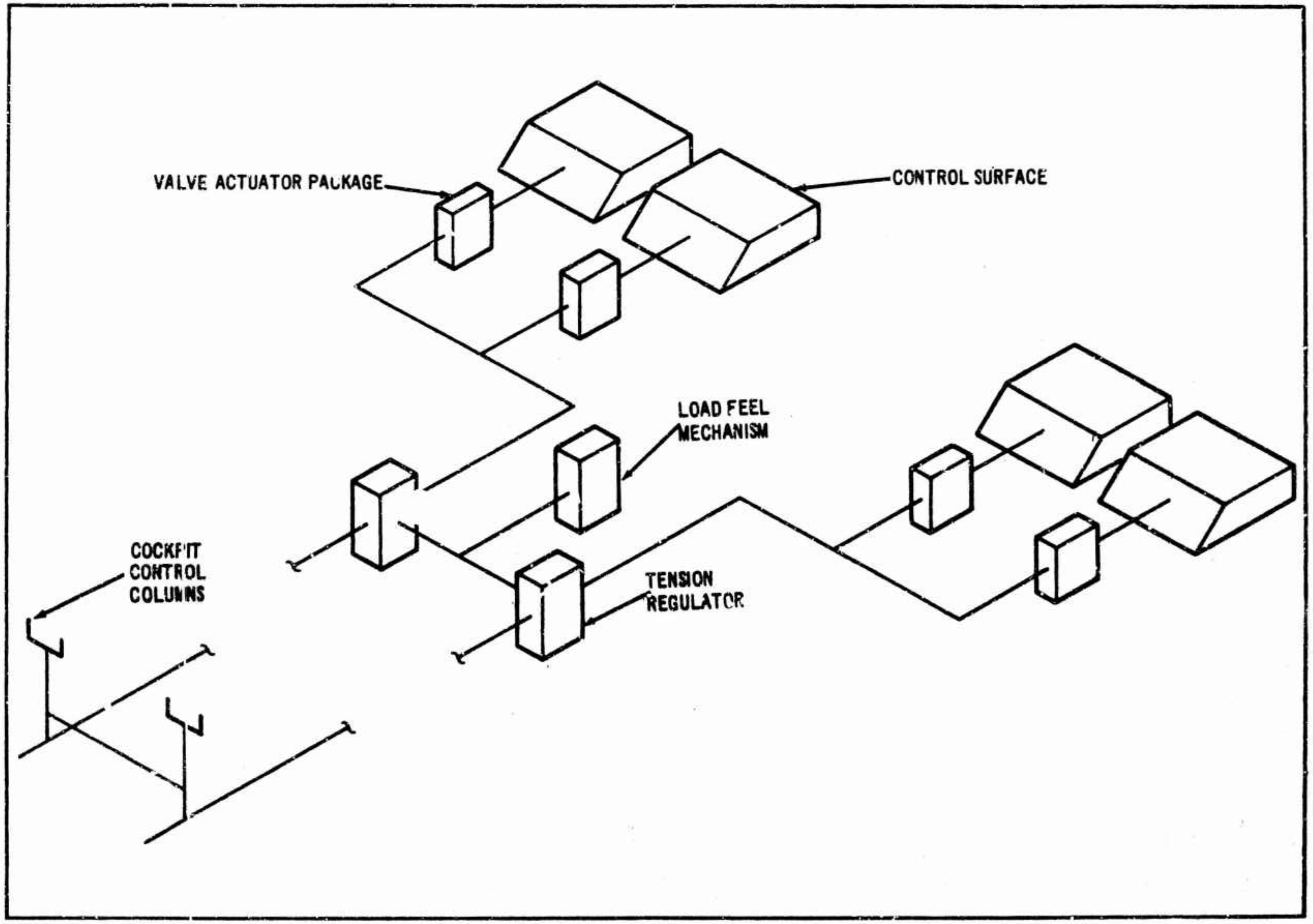

Figure 8. Elevator Mechanical Linkage Schematic

b. Dynamic Model

The time-dependent relationships which constitute the dynamic model are concerned solely with the dynamic performance of the actuation system. The development of a function to represent dynamic performance is complicated by the numerous quantities that must be considered to adequately describe it. Most measures of dynamic performance are stated in terms of the closed loop behavior of the system (for example, its bandwidth and rise time). It is particularly difficult to relate these terms to the independent variables used in the optimization procedure. It is clear that changing the terms involving the system pressure, actuator area, and actuator moment arm in the transfer function without changing the system compensation is meaningless. As a result, it was decided initially to use a method to derive a measure of dynamic performance which utilized control loop gains as well as the actuation system design parameters. The integrated-square-error (ISE) criterion was selected for this purpose because a system designed to this criterion results in an optimal following system. That is, the optimum system will be defined as the system that yields a minimum integrated-square error between a reference input and the system output (see Figure 9). Thus, the optimum system will provide the best reproduction of the input signal. In addition, the ISE was selected because it provided a numeric that described the optimum system and because a reference input having the desired bandwidth and rise time could be chosen.

Because dynamic performance was being optimized for only 10% input signals, it was possible to use a linear representation of the valve-actuator-load combination as shown by the actuation block of Figure 9. This linear model is developed in Appendix II.

The ISE can be calculated analytically and the optimal parameters evaluated for systems of low order. For this study, however, a two-level optimization procedure is required; first, the system independent variables must be specified and then, second, the parameters of the control system such as K_a (the amplifier gain) have to be evaluated. This results in nonlinear algebraic equations which are hard to solve; therefore, an alternate technique was employed because for the purpose of the study, only the ISE is needed.

The alternate technique consisted of simulating the system on the analog computer and evaluating the minimum ISE. This was done by first setting the parameters of the actuation system and then selecting the control system gains to yield the minimum ISE.

From the development of the linear model included in Appendix II, it is seen that a change in system pressure will affect only the K_q term. Because this term appears only in loop gains involving independent control system parameters, it is clear that the system can be made independent of pressure changes. Therefore, the only terms that can affect the system ISE are the actuator area and the actuator moment arm thus making it an implicit rather than an explicit function of the pressure losses through the plumbing and the metering valve. Further inspection of the linearized block diagram leads one to the conclusion that changes in area and moment arm should be identical

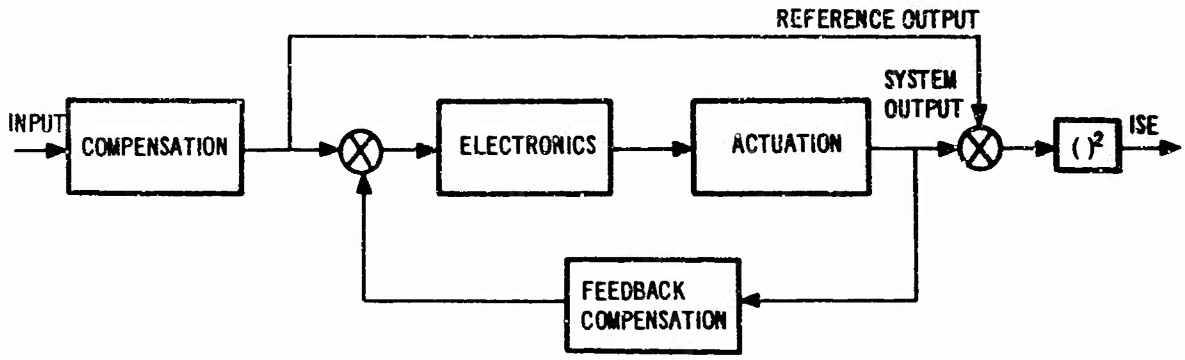

Figure 9. Integrated Square Error Generation

within a gain change in the forward loop. Simulation of the system on the computer verified this result and allowed the ISE to be expressed as a function of the product of area and moment arm. Because the actuator-load resonant frequency is also a function of this same product, it was found that ISE decreased when the product increased.

One disadvantage of the ISE is that it does not indicate when the dynamic performance is adequate. That is, because a following device is being designed, any increase in open-loop resonant frequency that allows more faithful following of the reference input will result in a decreased ISE. However, if the previous system were adequate, the improved performance indicated by the ISE is misleading because the improvement is not required. Another disadvantage of the ISE is that no knowledge is available of how well the system could have been made to perform if additional and/or different types of compensation had been used. For these reasons, it was felt that the ISE created an artificial function and did not contain adequate design information.

In an effort to define a new measure of dynamic performance that would be more meaningful (especially to the designer), a function of the open-loop resonant frequency was sought.

Experience at Douglas has shown that it is possible to achieve adequate performance of the control system if the load-actuator resonant frequency is at least four times the desired bandwidth of the actuation control loop. In some instances, it was possible to reduce this factor to as low as 2.5. Several things should be borne in mind concerning these statements: (1) the size of this factor is also a function of the flight control loop within which the actuation system is placed; (2) this work was done entirely with flow control valves and the load-actuator poles are virtually undamped in this case; and (3) factors greater than four allow the designer greater flexibility and simplify the problem of compensation. Figure 10 shows a plot of the open-loop poles and zeros of the system transfer function as a function of the load-actuator resonant frequency. The damping associated with these poles is caused by the dynamic pressure feedback. An additional feature of a function based only on the resonant frequency is that no compensation of the control loop need be considered as was the case with the ISE. Referencing the block diagrams in Appendix II, the load-actuation resonant frequency per actuator can be written as

$$\omega_n^2 = \frac{A_p R^2}{I} K_n = \frac{2\beta A_p^2 R^2}{I V_{an}}$$

The volume of the actuator can be written as

$$V_{an} = A_p Y_{PM} = A_p R \delta_{max}$$

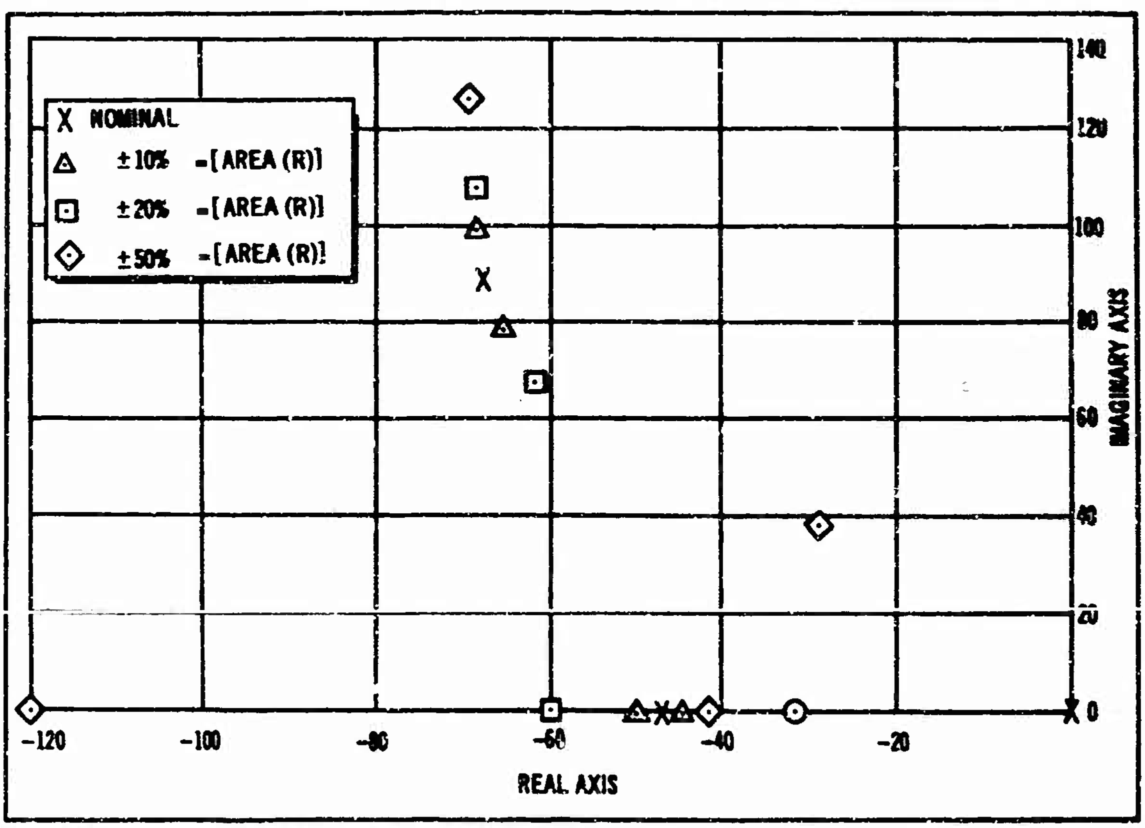

Figure 10. Resonant Frequency Root Locus

The resonant frequency becomes

$$\omega_n = \left[\frac{2\beta}{I\delta}_{max} A_p R\right]^{1/2}$$

Numerous functions can be developed if only ω_n is considered as a measure of dynamic performance. Two such functions are shown in Section IV.

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SECTION III

OPTIMIZATION TECHNIQUE

1. PROBLEM STATEMENT

The problem under consideration is the method for choosing optimal system parameters. The optimal system is defined as the system that maximizes the chosen performance index (P. I.). For this actuation design problem, the system pressure, actuator area, actuator moment arm, and tube diameters are chosen to maximize a function of the weight, volume, dollar cost, reliability, dynamic performance, and environmental sensitivity. To state this mathematically, let S be a physical system or process.

The parameter vector $\underline{u}$ is the system pressure, actuator area, actuator moment arm, and tube diameters and $\underline{x}$ is the output vector and is composed of the weight, volume, dollar cost, reliability, dynamic performance, and environmental sensitivity. Therefore, $P.I._{OPT}(\underline{u})$ must be defined. Let

$$P.I._{OPT}(\underline{u}) = \underset{\underline{u}}{MAX}\ P.I.(\underline{u})$$

$$P.I.(\underline{u}) = \sum_{i=1}^{n} f_i J_i(\underline{x})$$

$$x_j = g_j(\underline{u}) \qquad j = 1, 2, \ldots, m$$

In general, the equations defining the system will be nonlinear. The math model derived in the previous section related the parameter vector to the system outputs. The following sections relate the system outputs to the cost functions, J_i, and the cost functions to the total performance criterion.

2. COST FUNCTION DEVELOPMENT

The development of the cost functions is a task that requires considerable judgment by the designer. One fundamental problem is the generation of a numeric that will adequately represent the vehicle constraints. Also, it is desirable to have common bounds on all of the variables. If this is not done, the difference in the absolute values of the different system outputs will affect the optimization procedure. Therefore, a numeric bounded between zero and one was used for the cost functions. Thus, in a maximum searching system, one corresponds to the best possible system, and zero to the worst.

Development of the cost function to be used on each vehicle constraint will depend on the design requirements of that vehicle. For example, if all incremental decreases in system weight are equally important, a linear weighting on the total system weight might be desirable. One method of developing a linear weighting is to compare the calculated weight with a reference weight. For the study, the reference weight is defined as the weight calculated with use of standard design techniques. With use of this cost function, absolute changes from the reference weight (W_n) are weighted identically. One possible form of the cost function is

$$J_i = 1 - \frac{W}{2W_n}$$

Thus, when the weight is equal to the reference weight, $J_i = 1/2$, and varies as shown by Figure 11.

Note that the cost function could be less than zero if the weight increased to more than twice the reference weight. If it is meaningful to consider such cases for the system being designed, the cost function must be redefined.

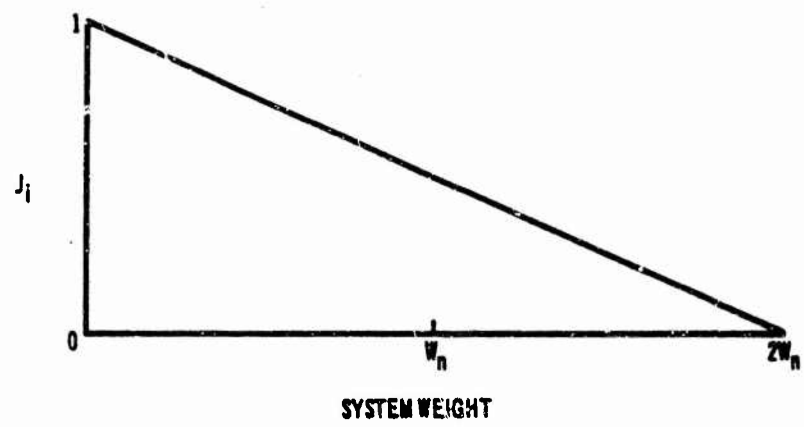

Figure 11. Weight Cost Function

For other systems, any actuation system weight less than a maximum allowable weight, ($W_{max.}$), might be permissible. For such a system, a cost function such as

$$J_i = 1/2 \left[1 - \text{sgn}\,(W - W_{max.}) \right]$$

might be appropriate. This cost function is zero when $W > W_{max.}$, and is one when $W < W_{max.}$. Beside.. the two possible cost functions mentioned, many interesting linear and nonlinear weightings could be derived to meet the requirements of a given design problem.

A vehicle constraint that requires special consideration is the volume of the system. Consideration of the elements that comprise an actuation system, leads to the conclusion that total volume is not a meaningful quantity for many of the elements. For instance, the length of the actuator is a more critical quantity than the total volume of the valve actuator assembly because the valve can be shaped to fit the space available. Therefore, volume per se was not used as the cost function. A quantity that is defined in this study as a space factor was substituted. The space factor is a combination of each element's critical dimensions, such as, actuator length, volume of the pump and accumulator, and the length and diameter of the reservoir. One method of deriving a space factor is to define a subcost function for each term of the space factor. The space factor is defined as the sum of these subcost functions. With use of the ideas presented earlier, a suitable cost function might be defined as follows: if the volume of the elements for which volume is the important factor is summed, a suitable subgoal might be

$$V_1 = 1 - Veq/2\,Veq_{nom}$$

where Veq is the current volume and Veq_{nom} is the reference volume which might be the value given by standard design techniques. If the length of the actuator is a critical dimension, a suitable subgoal might be

$$V_2 = 1 - ACTL/2ACTL_{nom}$$

where ACTL is the current actuator length and $ACTL_{nom}$ is a reference length. The space factor cost function might be defined as

$$J = 1 - VOL/2VOL_{nom}$$

where VOL is the current value of

$$VOL = \frac{1}{n \sum\limits_{i=1}^{n} f_i} \sum_{i=1}^{n} f_i V_i$$

and VOL_{nom} is a value calculated as a reference. The f_i's are emphasis factors which must be specified by the designer.

Each critical dimension should be assigned a subgoal and the weighting coefficients chosen to represent their importance to the total cost function.

The dollar cost function exhibited no unusual characteristic and was carried along as a linear cost function in this study.

The reliability cost function was found to be almost independent of the independent variables of the study and was simply carried along as a linear cost function.

The environmental sensitivity cost function is another factor that requires some special attention. It is clear that dynamic performance and environmental sensitivity are closely related. In terms of the actuator design problem, the only term in the environmental sensitivity cost function would be the bulk modulus of the oil. This term directly affects the actuator load resonant frequency, which is the term that affects the dynamic performance cost function. A definition of an environmental sensitivity cost function that might be meaningful is the comparison of the worst-case dynamic performance with the nominal dynamic performance for each design. Thus, if max. (Dyn. P.) is defined to be the maximum (or minimum) value of the dynamic performance cost function caused by the change in bulk modulus of the oil, then

$$J_i = 1 - \frac{\text{max. (Dyn. P.)} - \text{Dyn. P.}}{\text{Dyn. P.}}$$

This function could exceed the bounds of one if the maximum value of dynamic performance were greater than twice the nominal value. This would indicate extreme sensitivity to environmental factors and that the design should probably be rejected.

3. SYSTEM PERFORMANCE INDEX

As noted earlier, a performance index that can be used to indicate the total merit of a particular design is:

$$\text{P.I.} = \sum_{i=1}^{n} f_i J_i$$

where the J_i's again are the cost functions. The f_i terms are called emphasis coefficients and are used to denote the particular importance of each cost function to the design. If we make the following restriction

$$\sum_{i=1}^{n} f_i = 100$$

then the f_i's can be used to denote a percentage importance of each cost function to the whole system. It is apparent why the J_i's were bounded between zero and one; otherwise the emphasis coefficients would lose their connotation of percentage emphasis.

4. COMPUTER PROGRAMS

The computer programs developed for the optimal design problem are tools for the evaluation of the performance indexes. Two separate programs were developed to provide information to the designer.

The first program (No. 1) was developed primarily for providing information concerning the sensitivity of the cost functions. This is important for choosing the emphasis coefficients because the a priori guess may lead to a domination of the P.I. by one cost function. The maximum deviation of each cost function must be considered in context with its emphasis coefficient. To provide the sensitivity information, the parameter space was systematically searched by imposing a fixed grid on the space. The cost functions and the performance index were evaluated and printed at each point on the grid. This information can be evaluated and the emphasis coefficients updated as required. A flow diagram representing the essential features of this program is shown in Figure 12. A detailed description and a listing of the program are located in Appendix III.

The second program (No. 2) developed was primarily intended for choosing the optimal system. Numerous techniques of linear and nonlinear programming are available for this purpose. Because there are bounds on the parameter space and on the cost functions, the problem is fundamentally a nonlinear programming problem. One of the more popular searching techniques is the gradient method. Application of this technique to this problem, however, is complicated by the inclusion of the tube diameters as independent variables. The tube diameters are discrete variables; that is, they take on only a finite number of states, and special techniques must be used for handling discrete variables. Another popular searching technique is the random search or an extension, the adaptive random search. Its chief advantage is the simplicity of the programming. It has the disadvantage, however, that it may take excessive run time on the computer. There are two tradeoffs to consider: (1) whether it is more efficient, in terms of the computer, to calculate the gradient or to accept the wrong guesses of the random search, or (2) whether it is more efficient, in terms of manpower, to use a simple program at the expense of computer time. After some experience with the fixed-grid searching program, it was found that the program took little machine time, so the adaptive random search program was chosen. A flow diagram representing the essential features of this program is shown in Figure 13. Detailed descriptions of the random search technique and of the No. 2 program are located in Appendix IV. Listings that differ from the No. 1 program are also included.

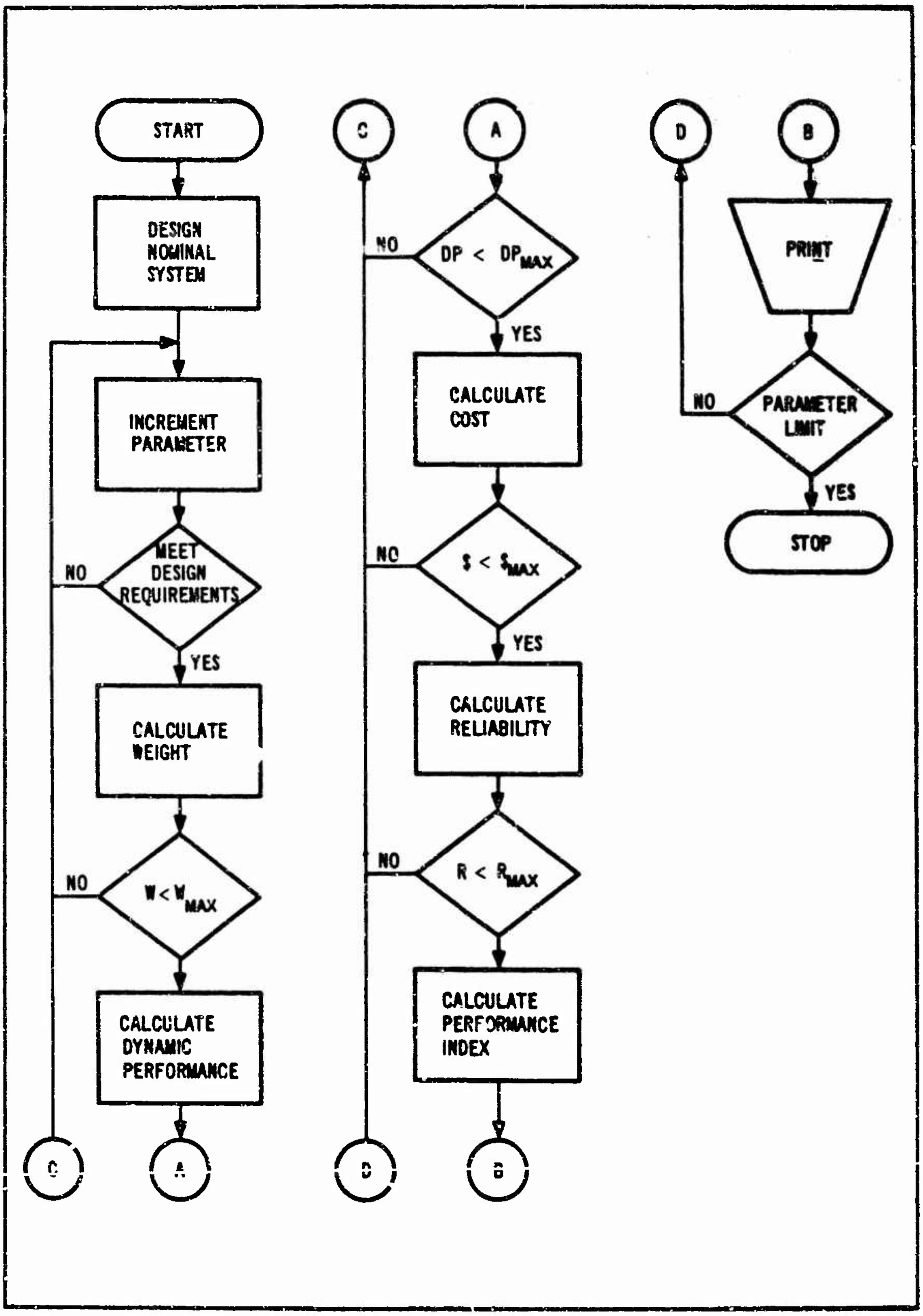

Figure 12. No. 1 Computer Flow Diagram

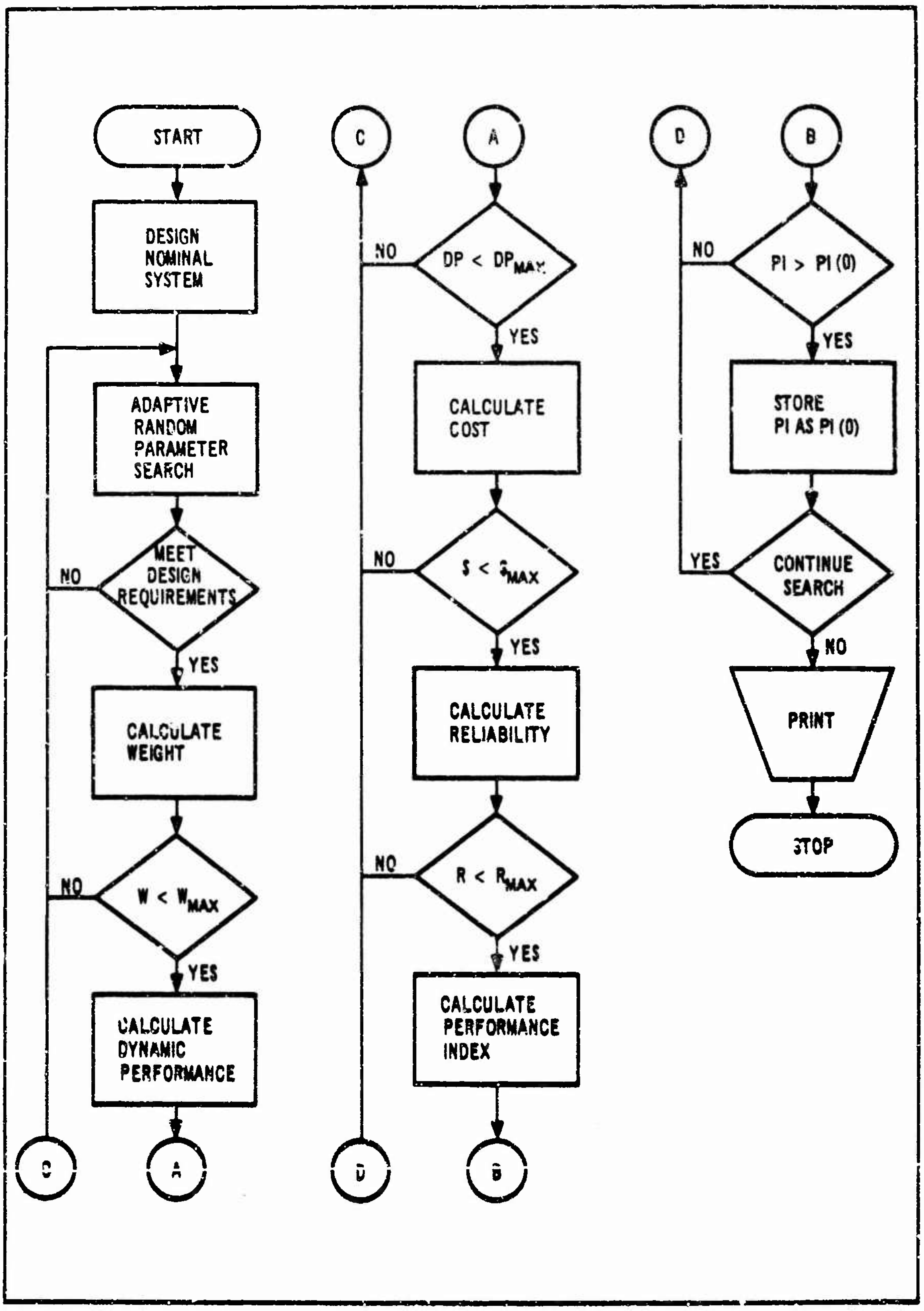

Figure 13. No. 2 Computer Flow Diagram

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SECTION IV

APPLICATION

The cost functions for the sample problem are developed in this section and the results of the optimization procedure are discussed. The design requirements and restrictions are given in paragraphs 2 and 3 of Section II. To supply a reference value for all cost functions using a linear weighting, a nominal system was designed using the design criterion of the Douglas proposal for the C-5A. This also provided a convenient measure of the merit of the new design.

1. COST FUNCTIONS

a. Weight Cost Function

In the derivation of the sample problem, it was felt that weight in each portion of the actuation system was equally important; in addition, it was felt that increases and decreases in the weight of each portion of the system were also equally important. Therefore, a cost function using the total weight of the system with a linear weighting was used. That is

$$J_i = 1 - W/2W_{nom}$$

When it was felt that weight in one section of the system was more important than weight in another section, a compound cost function, similar to the volume cost function described earlier, could be used. For example, weight that is located a considerable distance from the center-of-gravity (CG) of the aircraft might be more critical than weight located near the CG and, hence, the former weight should receive more emphasis in the cost function than the latter.

b. Space Factor Cost Function

With use of the ideas presented in Section III, the space factor cost function was defined as a sum of the critical dimensions of the elements of the actuation system. An equivalent volume was defined as the sum of the volumes of the elements for which total volume was thought to be an important factor. For this example, only the pump and the accumulator were included in this sum. Equivalent volume was added as one of the subgoals of the space factor cost function. The reservoir was not a shape that could be adequately described by a volume. It was thought that the diameter of the large end and the length of the reservoir were a better description. Therefore, these two critical dimensions were each included as subgoals. Another critical dimension was the length of the actuator. It was felt that volume was not an adequate descriptor because the valve can be shaped to fit the volume available. Length, however, can be very critical in terms of the design of the aircraft structure. For this problem, the diameter of the actuator was not critical.

In any aircraft with a narrow section, however, it might be very important. The original definition of the cost function included these four terms with equal weighting and with a linear subgoal for each term. The reference value for each of the linear subgoals was the value obtained using standard design practice. Thus, the cost function is written as

$$J_i = \frac{1}{4\sum_{i=1}^{4} f_i} \sum_{i=1}^{4} f_i Q_i$$

Where

$$Q_1 = 1 - \frac{Veq}{2Veq_{nom}}$$

$$Q_2 = 1 - \frac{ACTL}{2ACTL_{nom}}$$

$$Q_3 = 1 - \frac{RED}{2RED_{nom}}$$

$$Q_4 = 1 - \frac{REL}{2REL_{nom}}$$

In the design of the C-5A structure, the space available for the actuator was limited in length by the supporting structure. To include the effect of this design restriction, an additional weighting was placed on any design that required a length of actuator longer than the length readily available. That is, a term

$$1/8\ [1 - \text{sgn}\,(5.5 - R)]$$

was added to the actuator length subgoal, where 5.5 in. is the maximum length readily available. The term penalizes the use of any actuator longer than the maximum length on an equal basis. This restriction is not meant to limit the length, but to give some importance to the extra problems involved in using the longer moment arm.

c. Dynamic Performance Cost Function

As discussed in Appendix II, two variations were investigated. The ISE model indicates improved performance whenever the resonant frequency of the load-actuator combination is increased. This presents a problem, because system requirements already have been established. Investigation of the system on the analog computer showed that performance requirements of the system were met easily because of the high hinge-moment requirements. It was, therefore, felt that the ISE was misleading, because the system was not as critical as the ISE indicated. For this reason, plus the fact that designers are not normally familiar with ISE, an alternate definition of dynamic performance was selected. The alternate selection, which is in terms of system bandwidth, is

$$\text{TOTDYP} = 1 - \text{EXP}\,[-1/4(\omega_n - 0.1\omega)]$$

and is shown in Figure 14. TOTDYP is the output of the dynamic performance subroutine. The cost function is one minus TOTDYP.

There is a good reason for choosing this cost function. As stated in the math model of the system, experience shows that it is possible to compensate the system if the open loop poles are at least four times farther out in frequency than the desired bandwidth. The cost function places no penalty on the system if the resonant frequency is 10 times farther out in frequency than the desired bandwidth. As the resonant frequency decreases, an increasing penalty is placed on the design. Because the cost function could become negative for frequencies greater than 10 times the bandwidth, the function

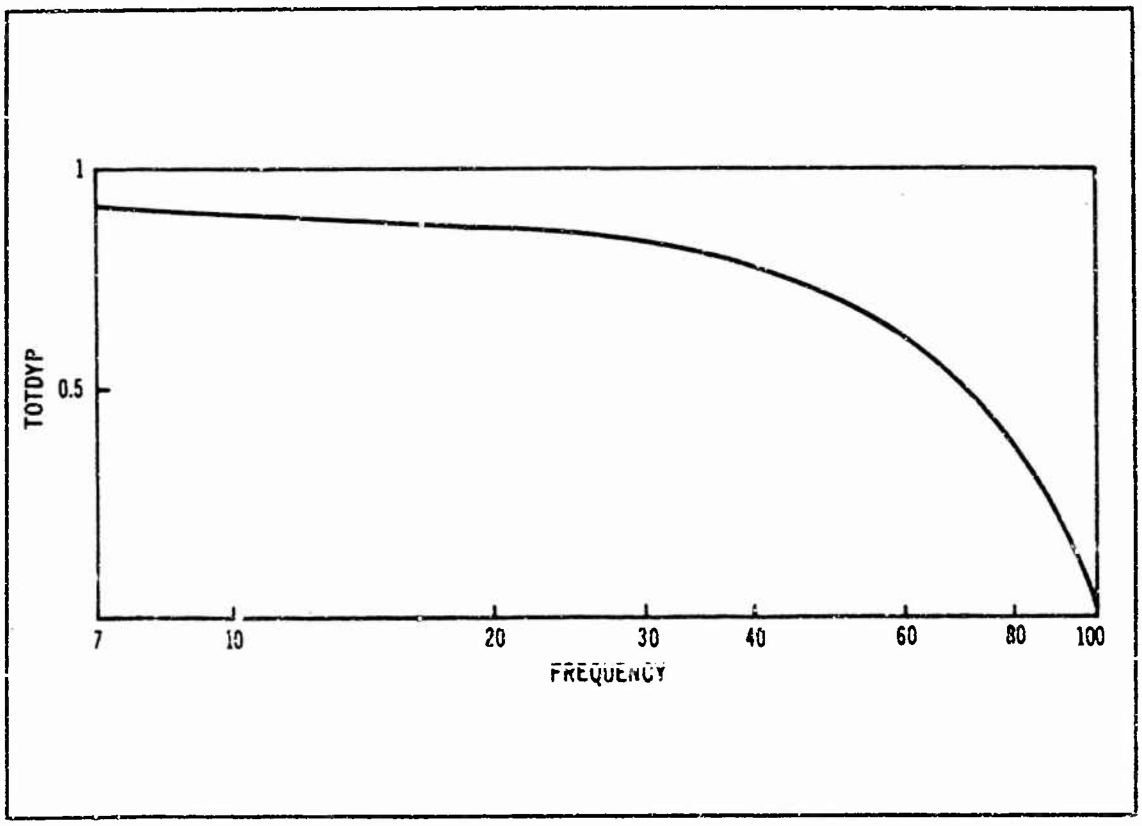

Figure 14. Dynamic Performance Function

was defined to be zero for any value of the exponential greater than one. This indicates that there is no particular value in having a wider bandwidth than required.

The cost function shown in Figure 14 may not be desirable because it heavily weights small decreases in resonant frequency. A more suitable cost function is shown in Figure 15.

This cost function weights small changes very lightly, but the weighting gets progressively heavier until it reaches the maximum at $4\omega_n$.

It is possible, in fact, that this excess bandwidth could be a problem in terms of bending modes or flutter frequencies of the aircraft. If certain frequencies must be avoided, it can be handled easily with this type of cost function by adding another term that takes a value of one at the frequencies in question. Such a term might be

$$\mathrm{EXP}\left[-(\omega_o - \omega)^2\right]$$

This term has the desired effect of heavily weighting frequencies close to the critical frequency and not weighting the other frequencies at all.

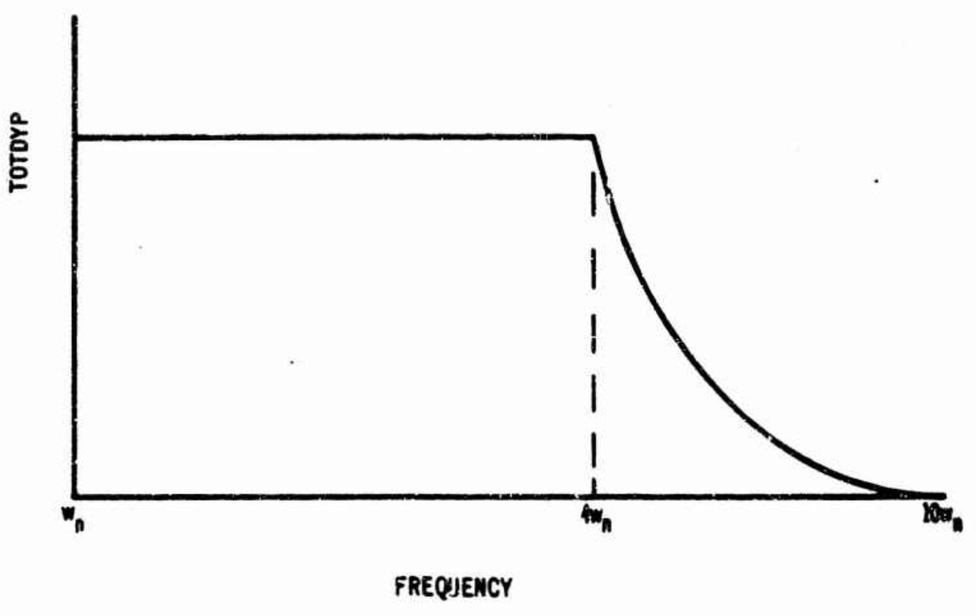

Figure 15. Alternate Dynamic Performance Function

One additional point should be noted; an additional refinement of dynamic performance could be made by including the functional relationship between system elasticity and the system pressure and area. For the example in this report, the effect was not significant because the resonant frequency was very high and the change in system elasticity is not great at the temperature and pressures involved. In problems where the resonant frequency is critical, this may be an important factor.

d. Environmental Sensitivity Cost Function

Because the elasticity of the system directly influences the resonant frequency of the system, the environmental sensitivity cost function is defined easily in terms of the dynamic-performance cost function. For the example problem, the bulk modulus of the oil and the elasticity of the actuator structure combine to change the elasticity of the system. However, it was found that this effect was not significant, especially because the dynamic performance was not a significant factor. For this reason, this cost function was not considered further.

e. Dollar Cost Function

The dollar cost function was taken to be an incremental cost and was not strongly a function of the independent variables. Because of its lack of significance, it was taken to be a linear-cost function and was assigned a small weighting.

f. Reliability Cost Function

Although it was felt that reliability would play a key role in the choice of an optimum system, sufficient data could not be found to establish a connection between reliability and the independent parameters. The cost function was carried along as a linear function with small emphasis, in the event that such a significance could be found.

2. EMPHASIS COEFFICIENTS

The relative importance of the various vehicle constraints is incorporated into the optimum design by selection of the emphasis coefficients. In this particular problem, weight was considered to be the most important factor. Dynamic performance and volume were given equal ratings followed by cost and then reliability. Not only should the relative importance be considered when selecting these coefficients but the extent to which the corresponding analytical relationships are developed and the accuracy or confidence in the basic input data should also be taken into consideration. The percentage values selected are as follows:

1. $f_{weight} = 30\%$.
2. $f_{dynamic\ performance} = 25\%$.
3. $f_{volume} = 25\%$.

4. $f_{cost} = 15\%$.

5. $f_{reliability} = 5\%$.

3. OPTIMIZATION PROCESS

With the cost functions defined and the model defined as in Section 2, the problem was put on the digital computer for evaluation of the optimum system. The computer program using the fixed-grid searching technique was used to generate data that could be used by the designer in evaluating the results. The designer has to know which cost functions are the most dependent on the independent variables. This is important for the proper choice of the emphasis coefficients and for an understanding of the results of the optimization process. The program pointed up the problem in the use of ISE. It was found that only the ISE showed a significant change with the independent variables and that the dynamic-performance cost function dominated the performance index. A review of this problem led to a redefinition of the dynamic-performance cost function which is felt to more adequately describe the physical situation.

All of the other cost functions showed little variation with changes in the independent variables. As can be seen in Figures 16 through 21, the graphs have large nominal values and show little curvature. Figure 16 shows a graph of weight as a function of pressure for a constant-torque system; that

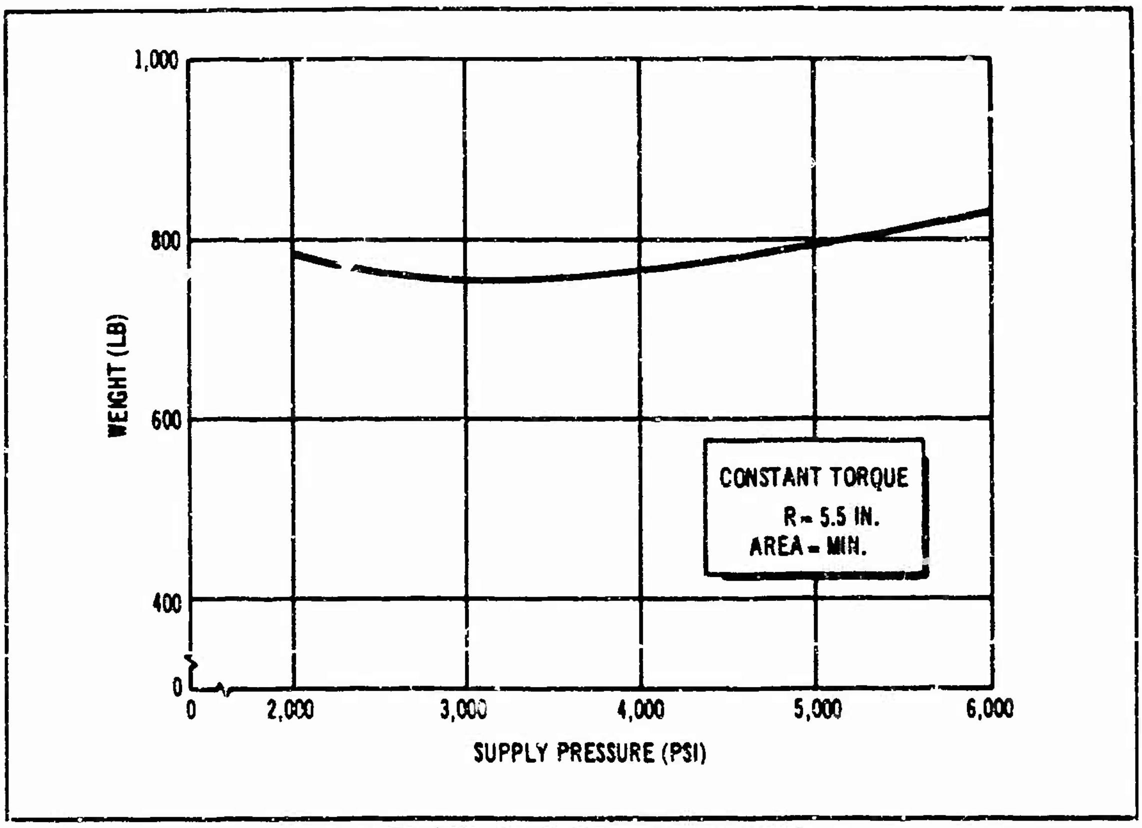

Figure 16. Weight as a Function of Pressure

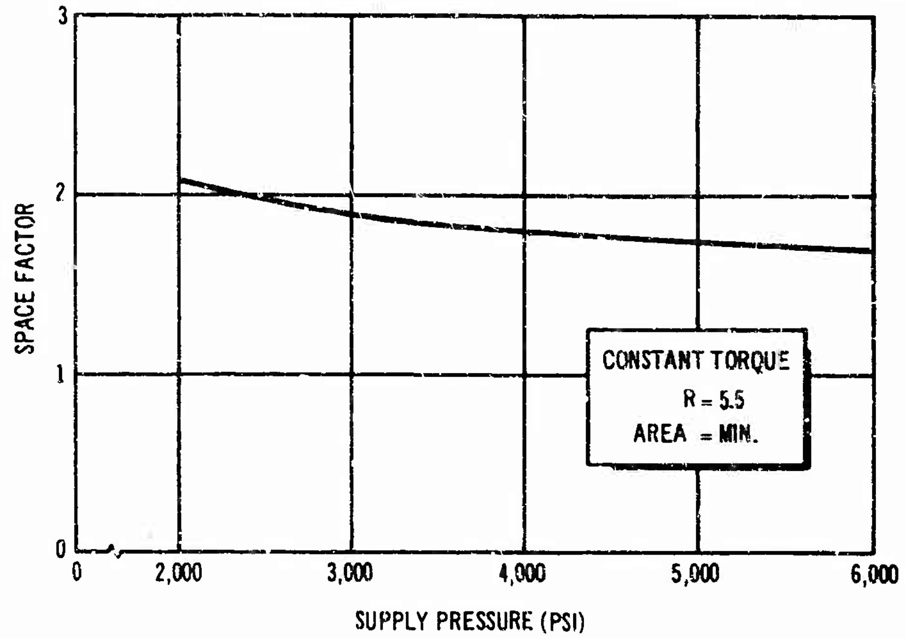

Figure 17. Space Factor as a Function of Pressure

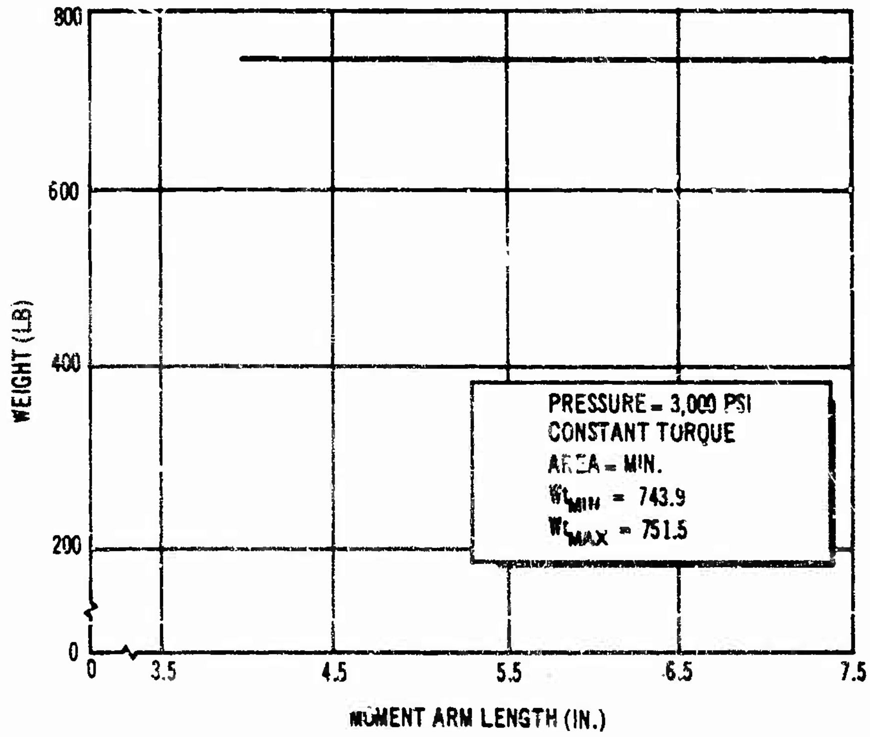

Figure 18. Weight as a Function of Moment Arm

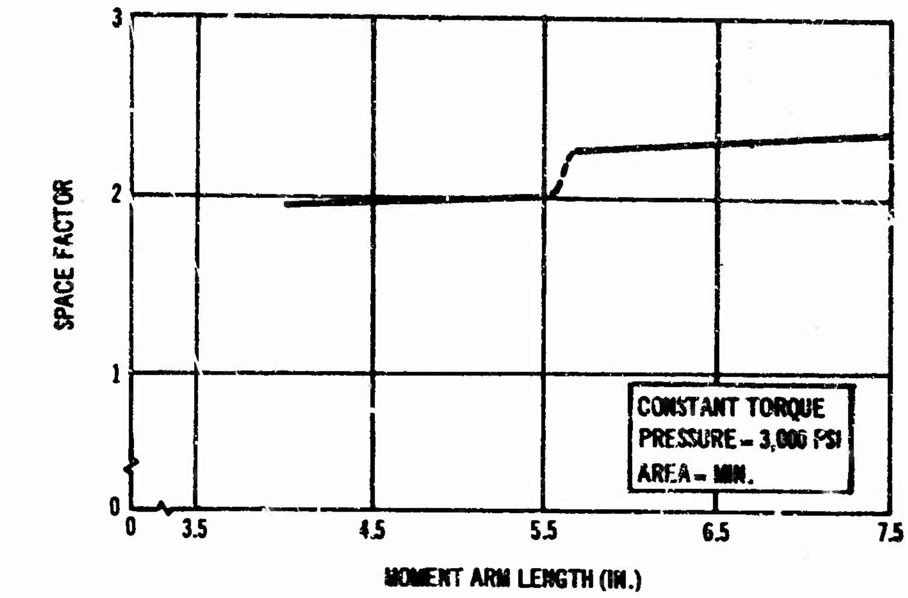

Figure 19. Space Factor as a Function of Moment Arm

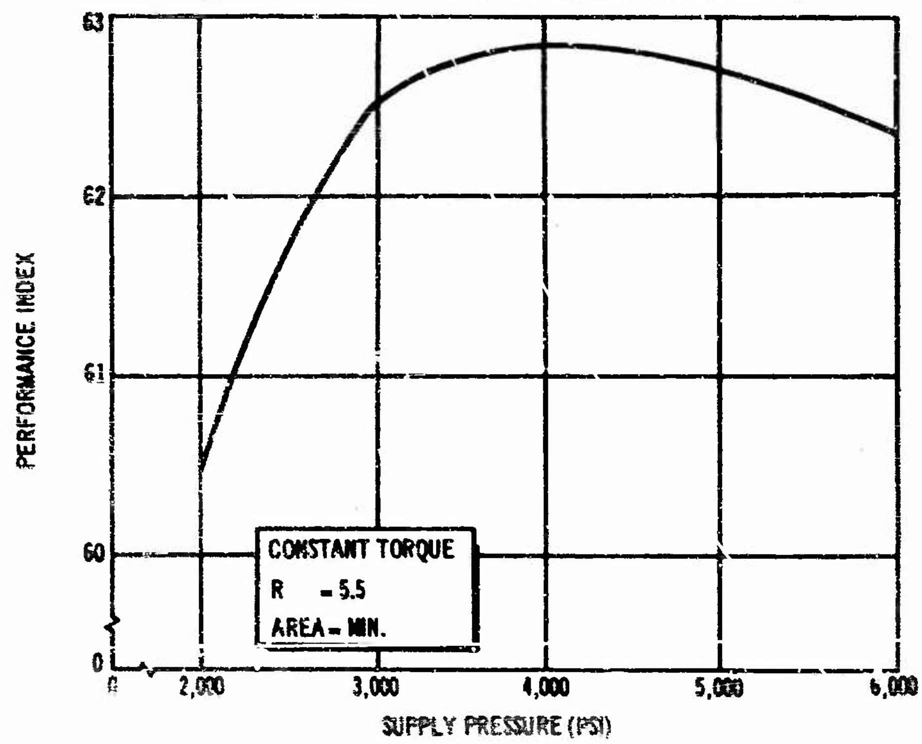

Figure 20. P.I. as a Function of Pressure

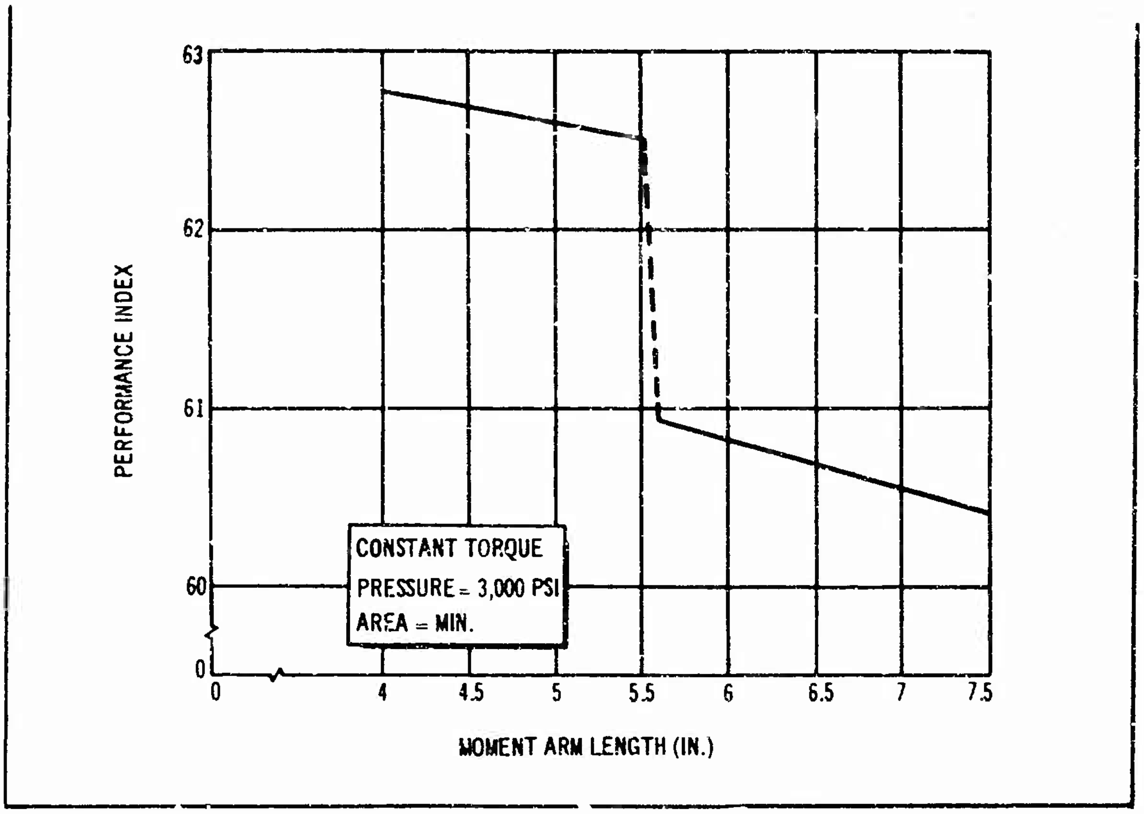

Figure 21. P.I. as a Function of Moment Arm

is, at each pressure, the minimum actuator area needed to supply the hinge moment and control surface rate for the given moment arm is used. The minimum area is determined by calculating that flow rate and corresponding actuator ΔP necessary to meet requirements taking into consideration the pressure losses through the plumbing and the metering valve. This curve shows a slight decrease as pressure increases from 2,000 to 3,000 psi. This decrease is primarily because of the smaller items required at the higher pressures. Above 3,000 psi, this decrease is lost in the increase in weight caused by the weight penalty to maintain the required safety factor. The minimum weight occurs at 3,000 psi and is 745 lb. Note that this minimum at 3,000 psi is somewhat artificial because of the weight penalty above 3,000 psi and the fact that below 3,000 psi, standard 3,000 psi equipment is used. If the 2,000 psi equipment were redesigned to reflect the lowered stresses there might not be a minimum at 3,000 psi. This is not standard practice, however. Disregarding this fact, the curve shows that most of the actuation system weight is fixed and that the pressure can be chosen over a wide range with little penalty in weight. For example, the maximum deviation in weight between 2,000 and 6,000 psi is approximately 9.3%, or approximately 70 lb. It is probable that such a small difference would not be significant in a large aircraft system.

Figure 17 shows a graph of space factor as a function of pressure for constant torque and for a moment arm of 5.5 in. The graph shows that the space factor (before normalization) decreases monotonically from a high of 2.20 at 2,000 psi to a low of 1.69 at 6,000 psi. The space factor decreases because the equivalent volume is a function of flow which decreases as the pressure increases.

The dollar cost of the system varied from a high of $10,250 at 2,000 psi monotonically to a low of $10,034 at a pressure of 6,000 psi. The difference in these numbers is less than the confidence present in the values themselves. The maximum deviation is approximately 0.2%.

Reliability was found to be independent of pressure.

When the revised dynamic performance index was used, dynamic performance was found to be independent of pressure.

Figure 18 shows a graph of weight as a function of the moment arm at a pressure of 3,000 psi and at constant torque. This shows that weight is virtually independent of the moment arm--a maximum deviation of less than 0.2%. The graph of actuator weight as a function of the moment arm in Appendix I (Figure 41) shows marked curvature, but the difference in scales is important. One point that might influence the weight as a function of moment arm that was not included, is the increased weight of the supporting structure as the arm decreases. The support was considered a rigid body in the study.

The space factor as a function of moment arm at a constant torque for a pressure of 3,000 psi is shown in Figure 19. The discontinuity reflects the additional weighting for moment arms that exceed the length that will easily fit in the available space. Again, the space factor has not yet been normalized.

The graphs of dollar cost and reliability as a function of moment arm are not shown because they are not significant. Dynamic performance is not shown because it is always acceptable.

The curves showing independent increases in the actuator area are not shown because all of the cost functions showed poorer performance. These data were taken at fixed pressure and moment arm and, thus, reflected a system of higher hinge-moment capability.

Figures 20 and 21 show graphs of performance index as a function of pressure and moment arm. The P.I. as a function of pressure peaks at 4,000 psi. The P.I. as a function of moment arm is dominated by the space factor cost function and indicates the choice of a short moment arm. The effect of changing the tubing sizes is to vertically shift the curves; the smaller the tube size, the larger the performance index.

The maximum P.I. is found at 4,000 psi, a 4-in. moment arm, minimum area, and all of the smallest possible tube sizes. At the maximum, the value of the P.I. was 67.49, the total weight was 638.9 lb, and the space factor was 1.57. Comparison with the nominal design (constant torque, 3,000 psi, 5.5 in. moment arm and largest possible tube sizes) indicates a weight savings of approximately 105 lb and a decrease in space factor of 0.43. An additional facet of decreased tube sizes that should be noted is that many of the low pressures are eliminated by the flow requirement and plumbing pressure losses at -40°F.

Referenced to the maximum P.I. using all the larger tube sizes (1 in. O.D. in Section 1 and 3/4 in. O.D. in Section 2) there is a possible weight savings of approximately 130 lb and a decrease in the space factor of 0.25 by using the smaller tube sizes (3/4 in. O.D. in Section 1 and 5/8 in. O.D. in Section 2). At these tube sizes, the plumbing pressure loss was 740 psi for a rated flow of 26 gpm at a fluid temperature of 120°F. Use of even smaller tube sizes was not possible because of the -40°F flow requirement. The above savings can be realized if proper valve operation is possible at lower than normal valve pressure drops.

Most of the results presented are reasonable, from the point of design practice. One possible disturbing item is the increase in P.I. as the moment arm decreases. Possibly, if the weight of the supporting structure had been included, this effect would not have been seen. Also, a short moment arm increases the effect of valve-actuator nonlinearities on control surface motion. To include the latter it would be necessary to make dynamic performance a compound cost function, like the space factor.

In an effort to gain additional insight into the nature of the function being maximized, data were taken along constant pressure, area, and moment arm planes. Also, these data were taken with the ISE dynamic performance cost function to show what the curves might be if dynamic performance was a critical factor.

Figure 22 shows the graph of weight as a function of pressure. It was found that space factor, ISE, and cost were constant as a function of pressure, when area and moment arm were held constant.

Figure 23 shows how weight, ISE, space factor, and cost vary as a function of area for fixed pressure and moment arm. The abscissa should be interpreted as follows: (1) indicates the minimum area for the pressure and moment arm given, (2), (3), and (4) indicate increases of 0.5 in.^{2} of area to the minimum for each number. Note that ISE improves as all of the other cost functions decrease. Figure 24 shows the same information for changes in moment arm.

For comparison, the previous weight and cost data taken on a constant-torque basis and plotted as a function of pressure will be shown to the same scale as used by Figures 22 and 23. Figures 25 and 26 show these graphs. Note that the scales on the ordinate are broken to show the curvature. The reason the weight is not identical to Figure 16 is because the linkage weight (which is constant) had not yet been added to the program and only one valve actuator assembly was being considered.

A similar comparison of cost functions as a function of area is not possible because an increased area reflects an increased torque capability. A

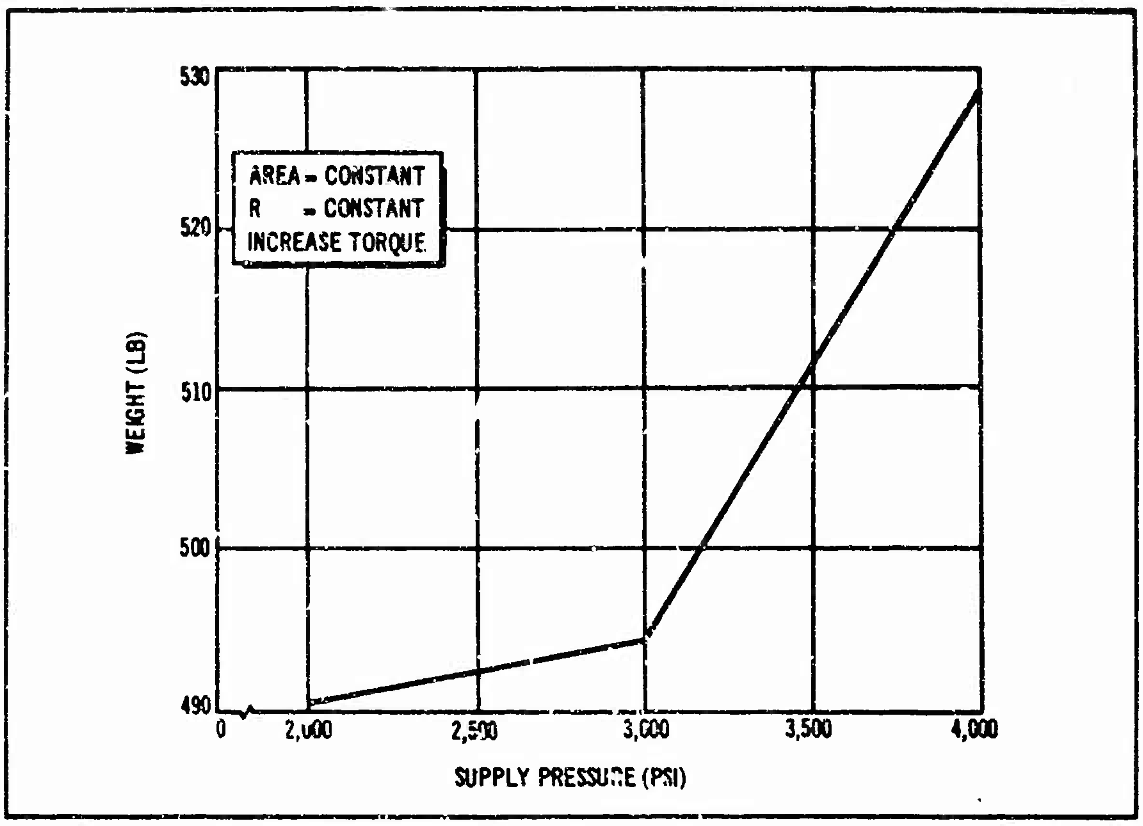

Figure 22. Weight as a Function of Pressure

comparison of graphs, showing changes in the cost functions as a function of moment arm, is not shown because the constant-torque curves contained no significant information.

The results of this work showed that there was little to be gained in terms of total performance by the choice of the independent parameters in this study.

4. RESULTS OF THE RANDOM-SEARCH TECHNIQUE

After the results of the fixed-grid search have been evaluated, it is possible to use the random-search routine. The fixed-grid routine provides information about the sensitivity of the cost functions necessary for the choice of emphasis coefficients. Once the emphasis coefficients have been chosen, application of the random-search routine is straightforward. For this particular problem, the random-search routine provided little information that was not readily available by looking at results of the fixed-grid routine. In some cases, however, the cost functions may vary a great deal as a function of the independent variables, and the cost functions may be difficult to interpret from the fixed-grid data. In these cases, the optimum system will be chosen by the random-search procedure. The search procedure could also give information about the sensitivity of the optimum, if all of the successful steps were printed. For the sample problem, it was found that the

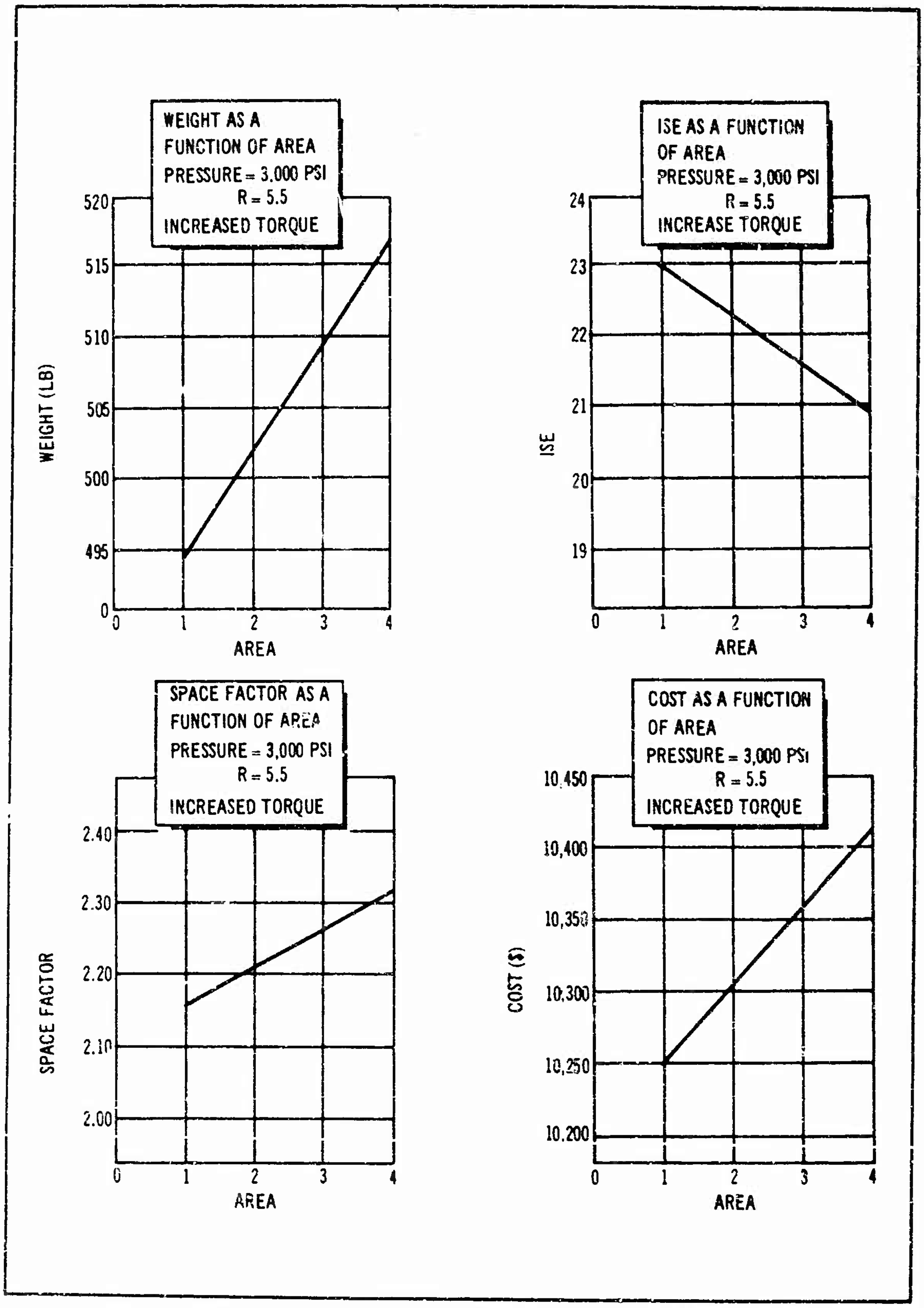

Figure 23. Weight, ISE, Size Factor, Cost as a Function of Area

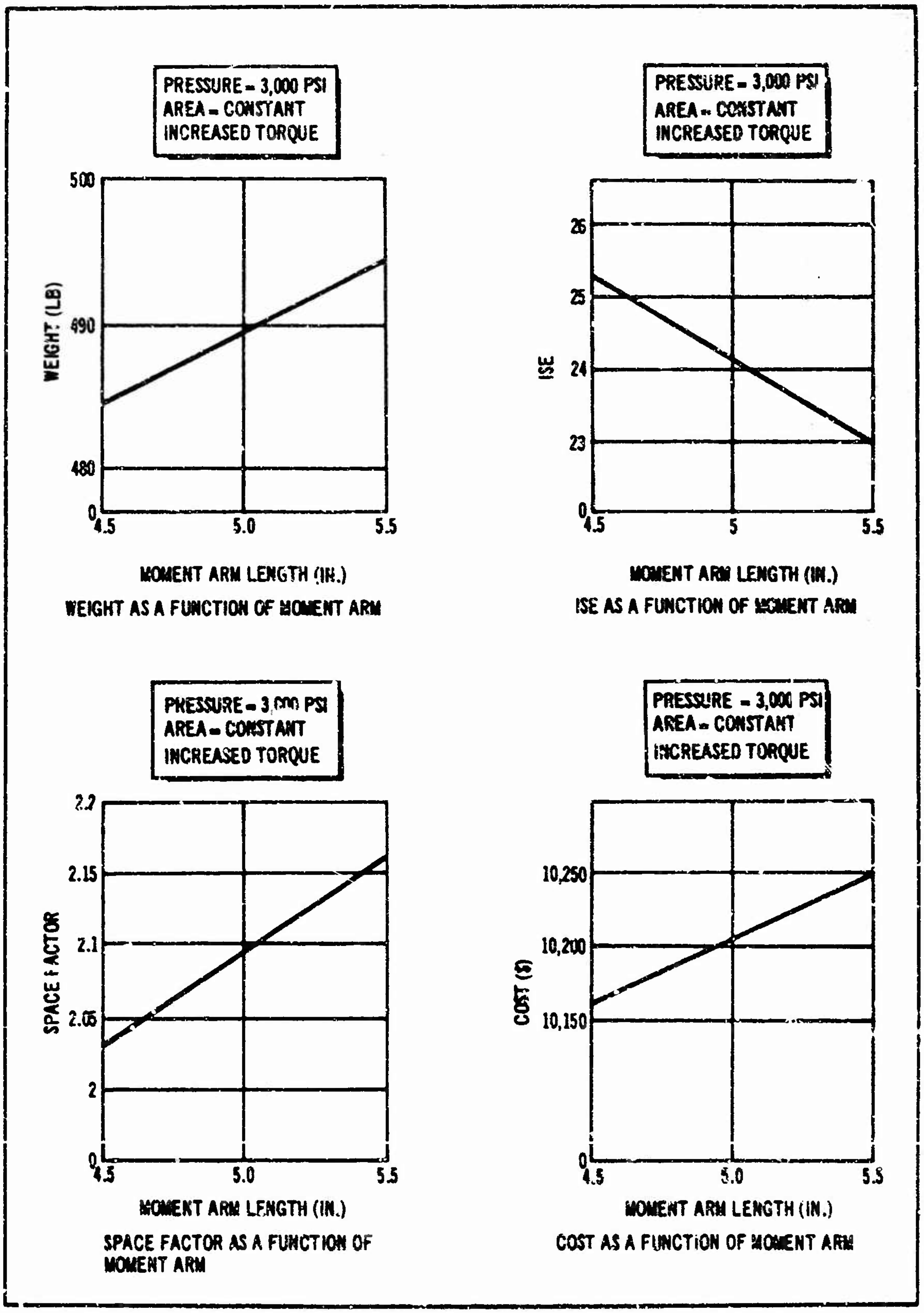

Figure 24. Weight, ISE, Space Factor, Cost as a Function of Moment Arm Length

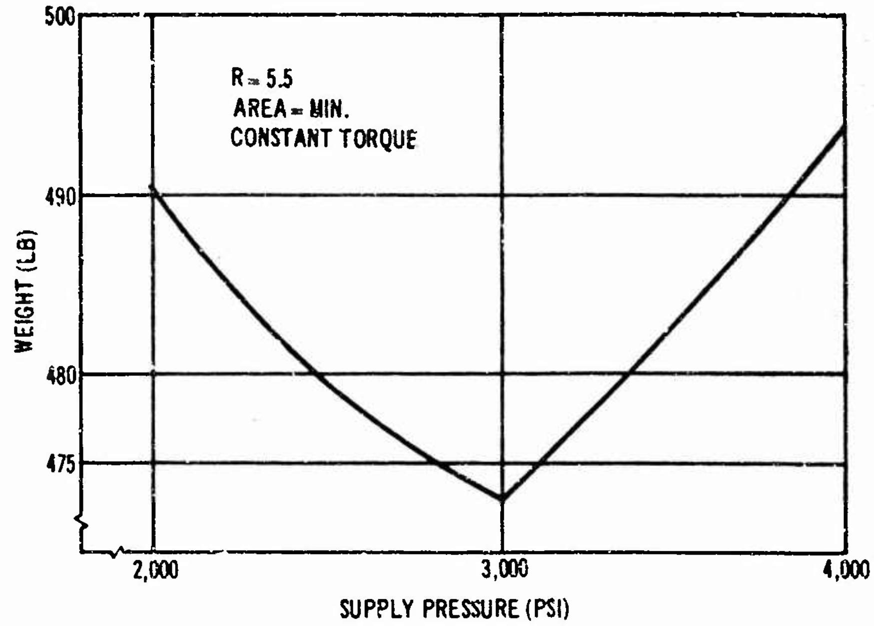

Figure 25. Weight as a Function of Pressure

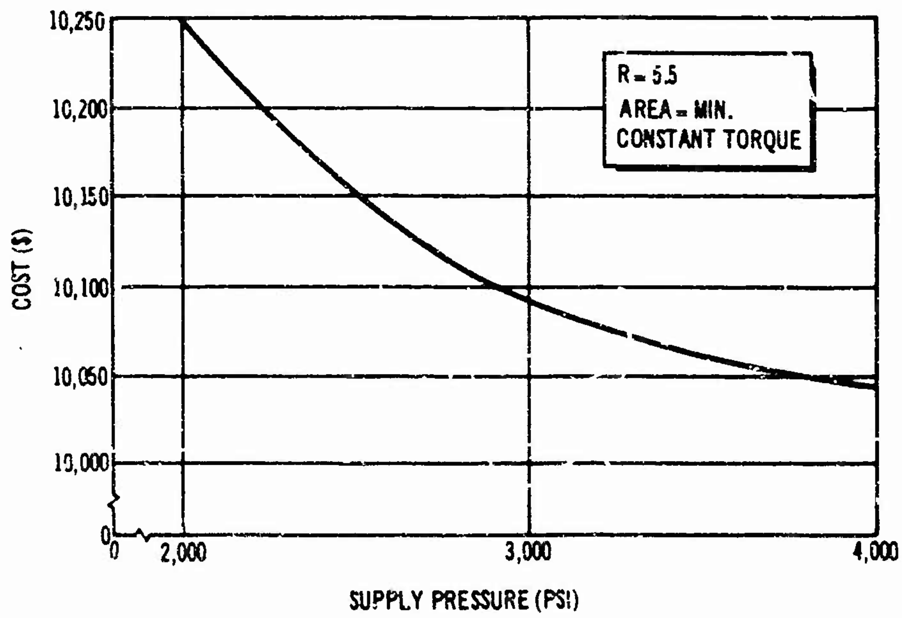

Figure 26. Cost as a Function of Pressure

random-search procedure took less time to reach the optimum than the fixed-grid routine. The search procedure was terminated when the computer made 1,000 unsuccessful steps in succession. One problem that led to the high number of unsuccessful trials was the choice of tube sizes. The performance index was maximized when all of the tube sizes were at their smallest value. Thus, the program had to choose all of the small tube sizes, which is a low-probability event (1/16), and also choose a set of parameters (pressure, area, and moment arm) that led to a larger performance index. The average run took less than 3 min. of computer time on the 7094. This time included approximately 1 min of compilation time.

The adaptive random-search routine showed little advantage for this particular example. The problem just cited, concerning the tube sizes, caused too many unsuccessful steps and negated the accumulation of knowledge gained by the successful steps. The adaptive search routine has been shown to be successful in reducing search time in other problems studied at Douglas (References 1 and 2).

SECTION V

EVALUATION

1. GENERAL

The main objectives of this study were to develop an optimal technique for the design of an actuation system, to determine guidelines for its use, and to evaluate the usefulness of the technique with respect to current design methods. The results indicate that an optimal technique, and its corresponding guidelines, have been developed. It appears, however, that the data obtained by application to the sample problem are not adequate to evaluate its merit. Detailed discussion of the optimal technique and corresponding guidelines is included in Sections III and IV. A detailed evaluation of its usefulness follows.

2. TECHNIQUE EVALUATION

Whether evaluating an optimal design technique or any new method, the questions that must be answered include the following:

a. Are the results better than those obtained by use of previous methods and, if so, can definite conclusions be made concerning the merit of the new method?

b. How universal is the new method, and are the results obtained by application to different systems directly comparable?

c. How does the cost in time and manpower compare between the old and new methods?

d. Are there related efforts being conducted elsewhere?

An evaluation by the review council provided answers that were most applicable to the first question because their primary function was a comparison between the optimal technique and previous design methods. In general, this evaluation emphasized the fact that the optimal design of the sample problem did not show much improvement over that obtained by using current design methods and as a result, it is not possible to draw definite conclusions on the merit of the optimal technique. The sample problem, a relatively elementary hydraulic system, was defined so as to expedite development of the technique, and so this is possibly a reason for the uneventful results. It was not apparent at the beginning of the study that more meaningful results would not be obtained. As it turned out, however, the relationships between selected design parameters and vehicle constraints were weak; so deviations from nominal values were relatively insignificant. If the restrictions and boundary conditions in the sample problem were relaxed and a greater number of interrelated parameters were considered, possibly more meaningful results would be obtained. Therefore, it follows logically that this technique should be studied further by conducting additional tests utilizing a complete actuation system.

Use of an optimal technique with a computer does not necessarily mean that an answer will be obtained that could not be obtained by using current design methods. On the other hand, it does mean that the monotonous repetition of conducting tradeoff studies between the various designs and design concepts can be largely eliminated. Also, by use of this new technique, more situations and parameter variations can be investigated, with the hopeful result that there is a more expedient method to find the optimal design. Even though this optimal design is selected automatically by application of the systematic searching routine (described previously), the results are still subject to the designer's background and experience through the mathematical model and other computer inputs. To standardize the optimal design technique, it is essential that model assumptions and constraint emphasis be controlled by the agency conducting the design investigation.

A possible use of this design technique is as a synthesis tool in an overall computer simulation of a complete vehicle flight-control system. By doing so, detailed actuation system design parameters, not normally included in the initial design studies, could be considered.

The second question is concerned with performance index comparison when optimizing designs have different basic concepts. Also, how universal is the technique itself when optimizing the same systems which have varying amounts of complexity and when optimizing systems which have differing design concepts? In other words, how much work is involved when converting this design method for use on different systems? Many such comparisons and conversions would occur when conducting tradeoff studies. Because only one system was investigated, additional studies will be required before these questions can be answered.

When an optimal design is determined, using this technique, the third question is: How much effort is involved and how does it compare with efforts required in current design methods? In an aerospace vehicle, where many design parameters are considered, use of the synthesis tool described in this report may well be worth the effort expended. These types of systems will have to be investigated further before an answer can be determined.

Actuation systems from other types of aerospace vehicles should be included in any such investigations. In high-performance missile systems, for example, dynamic response and power generation tradeoffs have a different order of importance and will have a different influence on the optimal design. As far as the optimization procedure is concerned, the primary effect of considering different aerospace vehicles will be in the choice of emphasis coefficients. Cost-function revisions should be secondary because of an attempt to build this generality into the basic program. Of course, a new set of equations relating the parameter vector to the output vector must be derived.

The fourth question is concerned with related efforts in the aerospace industry, of which the principal one is the study being conducted by General Electric, which has an objective similar to this study's. However, the work reported in this report deals with the optimization of an actuation system for a large transport aircraft, while General Electric's work (Air Force contract AF 33(615)-3587) is concerned with the optimization of actuation systems for a fighter aircraft. The results of General Electric's work have not yet been published.

The Martin Company conducted an actuation optimization program for NASA entitled, Optimization of Hydraulic Thrust Vector Control Systems for Launch Vehicles. A portion of this work is reported under NASA Accession Number N66-12611. An article was published (Reference 3) that describes the development of a computer program for related system cost investigations. Costs in this case included development cost, unit cost, weight cost, failures cost, development time costs, and others. Initially, optimization was obtained by observing trends in computer output data, rather than by application of some automatic optimization technique. It is possible, however, that work not yet reported may be investigating such techniques.

AiResearch Manufacturing Company, Division of the Garrett Corporation, conducted an optimization study for NASA on control moment gyros (CMG's). The CMG design was optimized with respect to a minimum of weight, size, bearing friction, and windage loss. Optimization was obtained by observing trends in computer output data. A report of this work is given in Reference 4.

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SECTION VI

CONCLUSIONS AND RECOMMENDATIONS

Based upon the preceding evidence, the following conclusions have been drawn:

1. A hydraulic actuation system that uses current design techniques provides a near-optimum design.

2. The optimal design is quite dependent upon the assumptions and conditions that accompany the design problem so the individual is still very much a part of the design process.

3. The design problem used in this study was beneficial in the development of the optimal technique but the ensuing results (because of the large nominal values and relative insensitivity of the cost functions) did not constitute an adequate base upon which to realize the full potential of the design technique.

4. It is possible that application would have more significance to high-performance missiles because variations in actuation-system performance have a decided effect upon flight-control-system performance. Also, in missile systems, it would be possible to conduct actuation power system tradeoffs.

5. It appears that the greatest value of optimal techniques applied to actuation-system design is connected with comparison of different design concepts for the same application. The study of various redundant configurations is a good example.

As a result, it is recommended that additional studies be conducted to obtain results that can be used to make a complete evaluation. This work should include:

1. Optimization of several actuation system design concepts for an actual piloted aircraft flight control system.

2. Conducting of similar studies on an aerospace vehicle other than a piloted aircraft.

These investigations would not only provide optimal designs for comparison with designs utilizing current methods, but would provide data that could be used to study the problems involved in the direct comparison of performance indexes. Also, the work involved in using the developed technique on different design concepts and different systems can be determined.

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APPENDIX I

STEADY-STATE MODEL RELATIONSHIPS

Each of the sections in this Appendix contains a summary of the parameters, output equations, and calculations used to generate the required information for the following (see Section II for further reference):

1. Pump.
2. Accumulator.
3. Reservoir.
4. Tubing.
5. Actuator.
6. Valve assembly.
7. Valve actuator package.
8. Fluid temperature.
9. System elasticity.
10. Fluid elasticity.
11. Actuator cylinder elasticity.
12. Mechanical linkage.

1. THE PUMP

Table I summarizes the parameter relationships associated with the pump. Derivations of these relationships follow the summary sheet.

a. Pump Weight

A Vickers slide rule can be used to determine the weight for various pump sizes. When plotted, the pump data can be approximated by a straight-line relationship as shown by Figure 27.

Based on the above data, pump weight as a function of pump flow can be expressed by

$$PUW = 7.5 + 0.615 \, (PUQ)$$

The system has four tandem valve-actuator packages and, so, has eight valve-actuation combinations. Assuming a 10% internal leakage loss, pump flow as a function of valve actuator flow can be expressed by

$$PUQ = 1.1 \, (8)(Q)$$

Combining this expression with the previous one, results in

$$PUW = 7.5 + 5.412 \, (Q)$$

Table I

PUMP SUMMARY SHEET

PAGE ______

NAME:	PUMP	SYMBOL	P U

INPUTS:	Q	OUTPUTS	P U W
	P R E S		P U C
			P U V
			P U R

OUTPUT EQUATIONS

WEIGHT	P U W (PRES-3000)>0	=	(7.5+5.412*Q)(1.+.00005(PRES-3000.))
COST	P U C	=	1400.+123.2*Q
SIZE	P U V	=	40.+199.7*Q
RELIABILITY	P U R	=	465.+.0111*PRES*Q
WEIGHT	P U W (PRES-3000)≤0	=	7.5+5.412*Q

NOTES: Data obtained from Vickers Incorporated, Division of Sperry Rand Corporation, Detroit, Michigan.

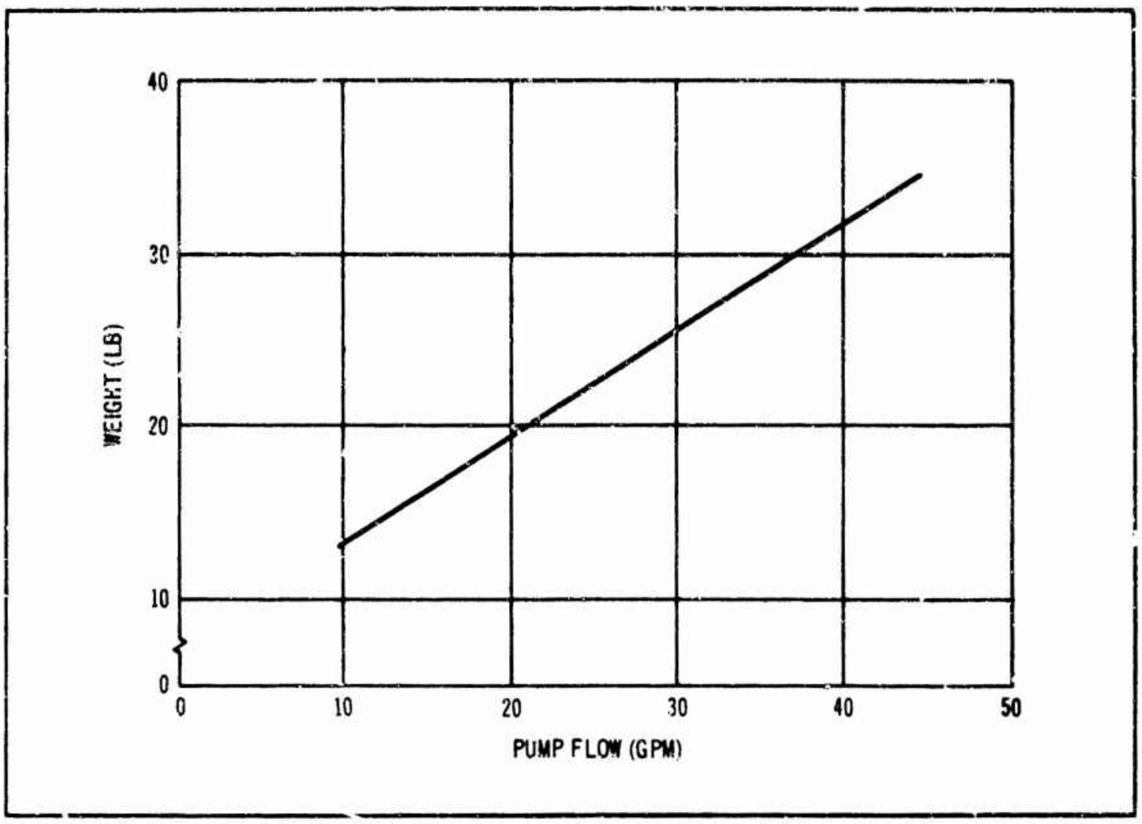


Figure 27. Pump Weight Data

Pump weight is also a function of system pressure. Data from Vickers indicate an approximate weight increase of 5% for an increase in pressure from 3,000 to 4,000 psi. If no weight reduction is assumed for pressures less than 3,000 psi and the same increase is assumed for every 1,000 psi above 3,000 psi, then the pump weight expressed as a function of valve flow and pressure is

$$PUW = [7.5 + 5.412\ (Q)]\ [1 + 5 \times 10^{-5}\ (PRES - 3,000)]$$

where

$$(PRES - 3,000) \geq 0$$

(It is assumed that the pressure denoted by PRES is a nominal pressure. As a result, it is not necessary to consider the pump return pressure of 70 psi.)

b. Pump Cost

The pump cost of the various sizes and for an arbitrary quantity can be approximated by the straight-line relationship shown by Figure 28.

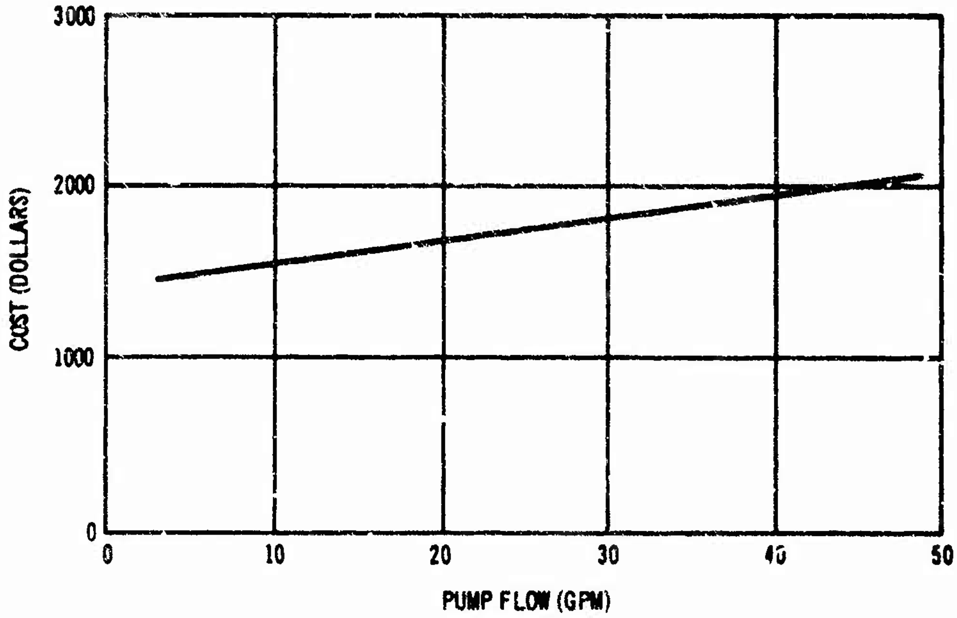


Figure 28. Pump Cost Data

From the above data, pump cost as a function of valve flow can be expressed by

$$PUC = 1,400 + 1[illegible]3.2\,(Q)$$

c. Pump Size

The size of the various Vickers pumps expressed as cubic inches of volume can be approximated by the straight-line relationship shown by Figure 29.

Based on the data shown in Figure 29, pump size as a function of valve flow can be expressed by

$$PUV = 40 + 199.7\,(Q)$$

d. Pump Reliability

The pump reliability is a function of the load so a smaller pump has a lower failure rate. If horsepower is used as a measure of the load, reliability can be correlated with flow and pressure. Based on the assumption that the failure rate is reduced about 10% between a Vickers 3915 and 3911 pump at a

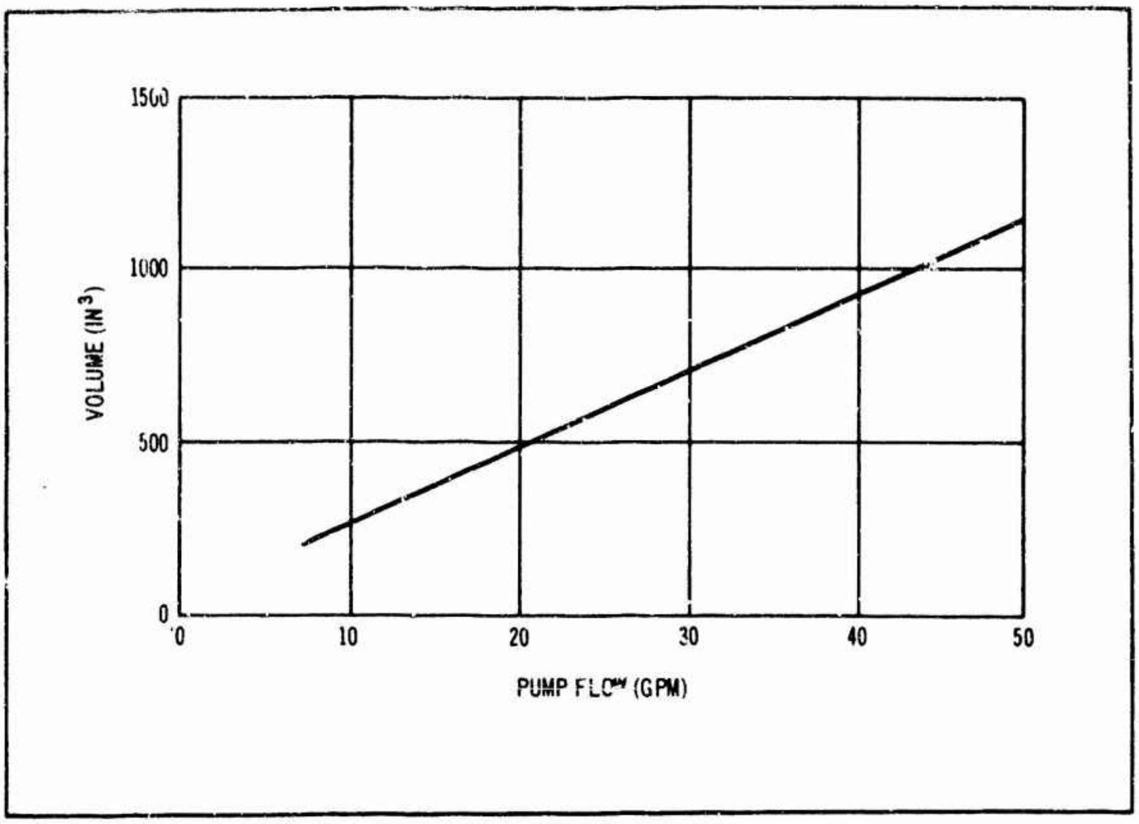


Figure 29. Pump Size Data

pressure of 3,000 psi, pump reliability expressed as the number of failures for 10^6 operating hours can be approximated by the straight-line relationship shown by Figure 30.

Based on the data shown in Figure 30, pump reliability as a function of pressure and valve flow can be expressed by

$$PUR = 465 + 0.0111 \text{ (PRES) (Q)}$$

2. THE ACCUMULATOR

Table II summarizes the parameter relationships associated with the accumulator. Derivations of these relationships follow the summary sheet.

An accumulator is used to damp out pressure surges caused primarily by sudden opening or closing of the metering valve and to smooth out the ripples in the hydraulics which are generated by the pump.

For this study, accumulator design will be based upon the additional flow capacity required to prevent system pressure from dropping below a specified value after a sudden valve opening and for the period of time the pump takes to increase its output flow from a minimum to maximum value. Pressure at this time is difficult to calculate accurately because the accumulator

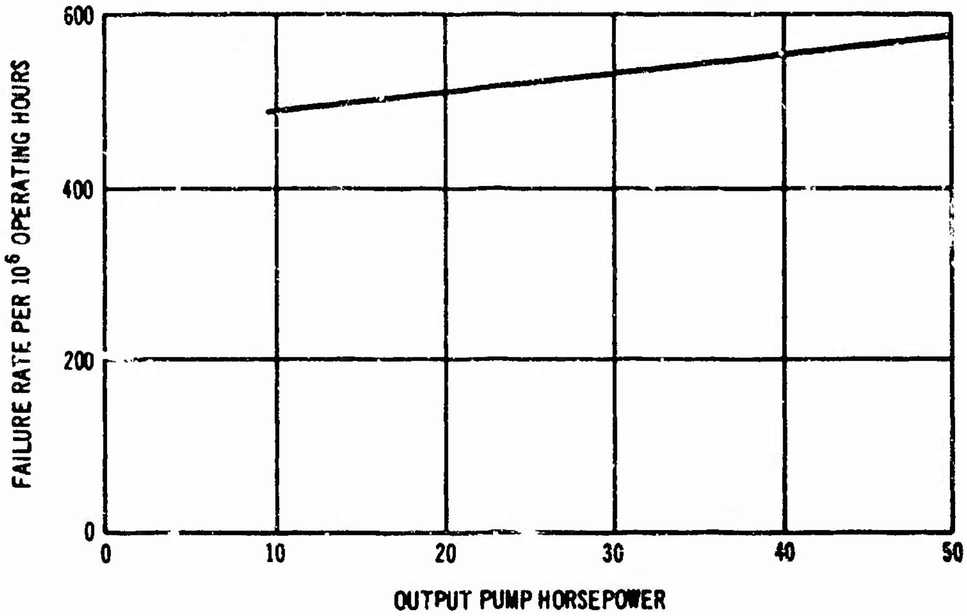


Figure 30. Pump Failure Rate

is discharging, which reduces pressure, and the pump flow is increasing, which increases the pressure. As a result, the minimum design pressure will be obtained simply by averaging initial system pressure with the resulting accumulator pressure obtained after the period of time that corresponds to the pump time constant. Charging or compression of the gas in the accumulator will be considered an isothermal process and discharging or expansion of the gas will be an adiabatic process.

The pump time constant, as shown by Figure 31, is only 0.06 sec. With a control surface rate requirement of 40°/sec, the control surface will travel less than 1° before the pump will be able to put out 100% flow when starting from 5% at a nominal speed. As a result, the minimum acceptable pressure when using flow from the accumulator will be assumed to be 2,400 psi. This is arbitrary and is set at this low pressure because the longest it can last is only 0.06 sec. The flow required from the accumulator when increasing the demand from 5% to 100% by stepping valve position is shown by the triangle on the previous sketch that is bounded by the pump flow and 100% flow lines. Assuming that 100% flow is about 30 gpm, the accumulator flow is

Table II

ACCUMULATOR SUMMARY SHEET

PAGE ______

NAME: ACCUMULATOR   SYMBOL A C C

INPUTS: P R E S   OUPUTS A C C W

A C C C

A C C V

A C C R

OUTPUT EQUATIONS

WEIGHT	A C C W	=	3.+(PRES-1000.)*.001
COST	A C C C	=	150.
SIZE	A C C V	=	90.
RELIABILITY	A C C R	=	12.3

NOTES: Data obtained from Bendix Pacific Division, Bendix Aviation Corporation, North Hollywood, California.

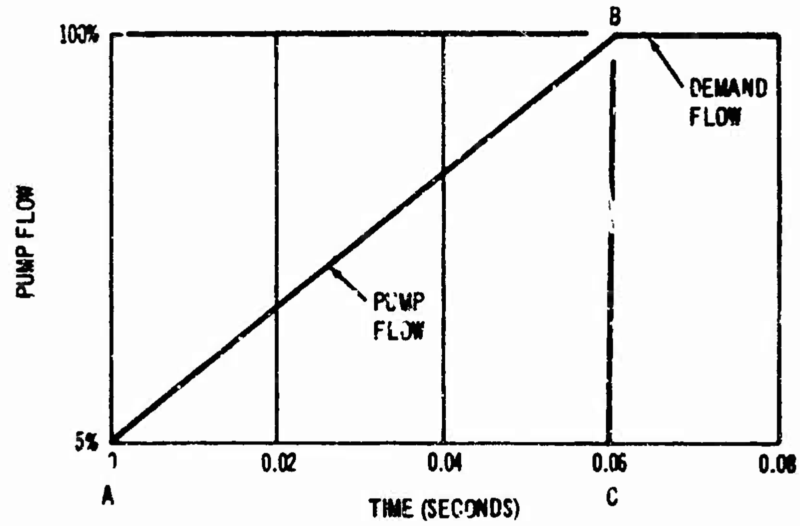


Figure 31. Pump Response Time

$$ACCQ = \text{Area of triangle ABC}$$

$$= \left(\frac{1}{2}\right)\left(\frac{30 \text{ gal}}{\text{min.}}\right)\left(0.06 \text{ sec}\right)\left(\frac{3.85 \text{ in}^3\text{-min.}}{\text{gal-sec}}\right)$$

$$= 3.47 \text{ in.}^3$$

With a 5-in. spherical accumulator precharged to 2,000 psi and an initial gas volume of 60 in.^{3}, an isothermal filling to 3,000 psi will result in a new gas volume of

$$P_1 V_1 = P_2 V_2$$

$$2{,}000 \ (60) = 3{,}000 \ (V_2)$$

$$V_2 = 40 \text{ in.}^3$$

If 3.47 in.^{3} of fluid is used, the resulting pressure following an adiabatic expansion will be

$$P_2 V_2^{1.4} = P_3 V_3^{1.4}$$

$$P_3 = 3,000 \left(\frac{40}{43.47}\right)^{1.4}$$

$$P_3 = 2,670 \text{ psi}$$

The minimum calculated pressure would be, therefore,

$$P_{min.} = \frac{3,000 + 2,670}{2}$$

$$= 2,835 \text{ psi}$$

which is well above the minimum allowable pressure. This particular accumulator also can be used with systems of larger flow capacity. Working backwards, starting with a minimum allowable pressure of 2,400 psi and maintaining a 2,000 psi precharge, 100% flow can be as high as 153 gpm.

This type of accumulator is quite flexible and is, therefore, independent of system flow. As a result, its cost, size, and reliability will be assumed constant and will have the following values:

Cost - $	ACCC = 150.
Size - in.^{3}	ACCV = 90.
Reliability - failures per 10^6 operating hours	ACCR = 12.3

The weight, however, is a function of pressure and can be approximated by the straight-line relationship shown by Figure 32.

Based on the above data, accumulator weight as a function of supply pressure can be expressed by

$$ACCW = 3 + (PRES - 1,000)(10^{-3}).$$

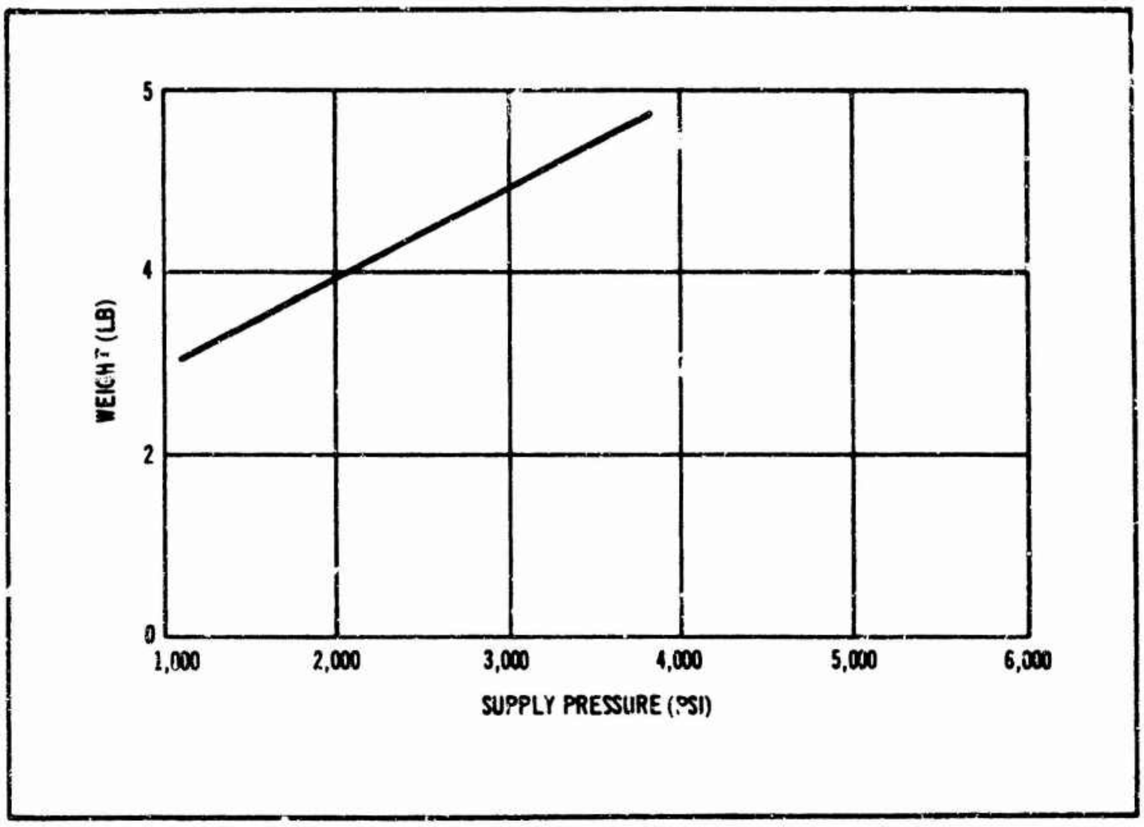


Figure 32. Accumulator Weight Data

3. THE RESERVOIR

Table III summarizes the parameter relationships associated with the reservoir. Derivations of these relationships follow the summary sheet.

a. Reservoir Weight

The reservoir size is dependent on the quantity of fluid in the system. Because the tubing contains most of the fluid, reservoir design is primarily a function of tube size. Weight, therefore, can be calculated by a standard Douglas calculation used for preliminary sizing, as follows:

1. Assume a tubing fluid weight (TUFLW) and then add a number of factors.

2. Add thermal expansion factor = 5.5% (TUFLW).

3. Add leakage factor = 5% (TUFLW).

4. Add accumulator fluid weight = ACCFLW.

TABLE III
RESERVOIR SUMMARY SHEET

PAGE ______

NAME: RESERVOIR

SYMBOL R E

INPUTS: I U F L W

OUPUTS:
R E W
R E C
R E L
R E D
R E R

OUTPUT EQUATIONS

WEIGHT	R E W	= 1.8+.219*TUFLW
COST	R E C	= 245.+.91*TUFLW
SIZE	*	
RELIABILITY	R E R	= 16.
*Length	R E L	= 9.+.18*TUFLW
*Diameter	R E D	= 6.+.04*TUFLW

NOTES: Data obtained from Aircraft Division, Douglas Aircraft Co., Long Beach, California.

5. Total these fluids weights and add more factors.

The sum of these weights is:

Total = TUFLW + 5.5% (TUFLW) + 5% (TUFLW) + ACCFLW.

6. Add thermal expansion factor again = 5.5% (total).

7. Add leakage factor again = 5% (total).

8. Add accumulator fluid weight again = ACCFLW.

9. The total of 6, 7, and 8 is the reservoir fluid weight, so

REFLW = 5.5% (total) + 5% (total) + ACCFLW.

10. Reservoir hardware weight is 85.5% of its fluid weight, so

REW = REFLW (1 + 0.855).

Assuming a nominal accumulator fluid weight of 1 lb, the reservoir weight can be approximated by the following straight-line relationship shown by Figure 33.

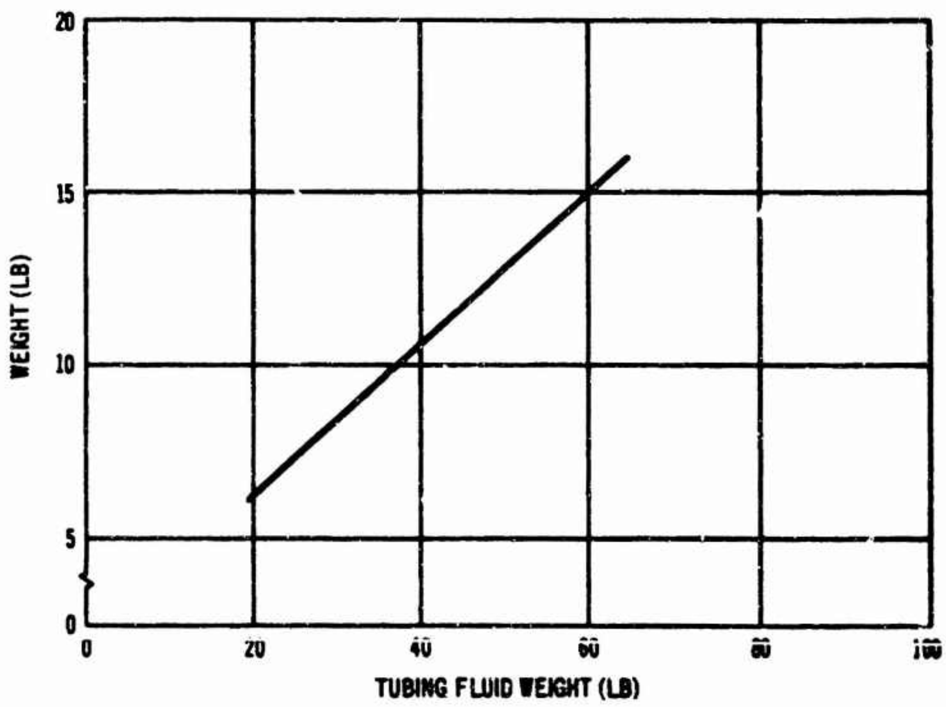


Figure 33. Reservoir Weight Data

Based on the above data, reservoir weight as a function of tubing fluid weight can be expressed by

$$REW = 1.8 + 0.219\ (TUFLW)$$

Because return pressure will not vary appreciably and because the major portion of the reservoir is concerned with return pressure, it will be assumed that the reservoir constraints are not functions of the supply pressure.

b. Reservoir Cost

The variation in cost will depend on the change in the amount of material required because the same configuration will be used in each case. Even though this change is minor, the relationship shown in Figure 34 will be used to determine reservoir cost.

Based on the data shown in Figure 34, reservoir cost as a function of tubing fluid weight can be expressed by:

$$REC = 245 + 0.91\ (TUFLW)$$

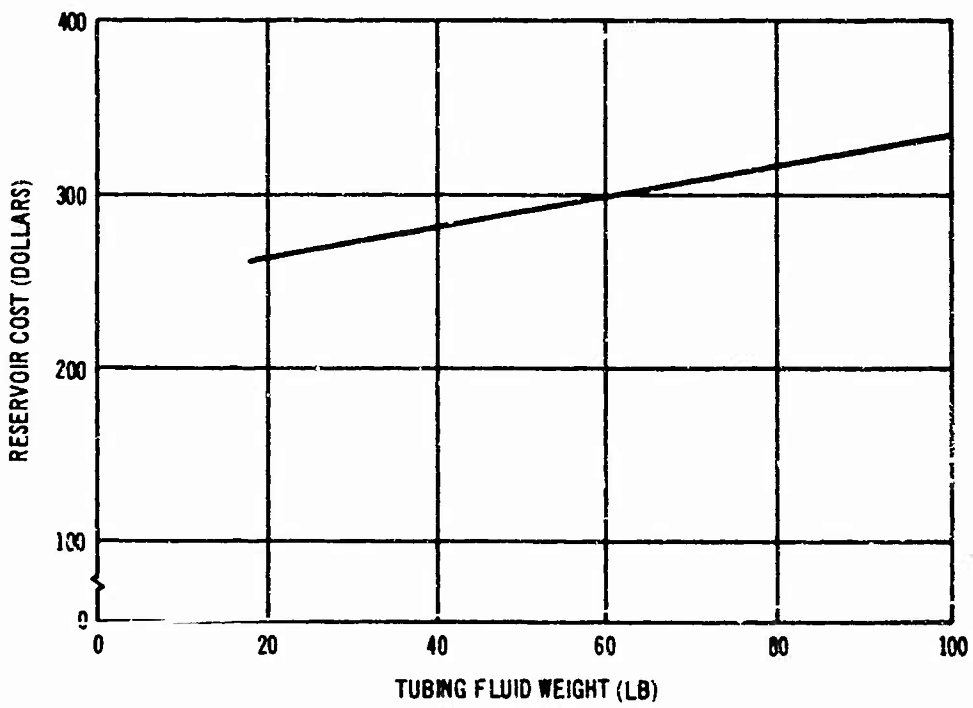


Figure 34. Reservoir Cost Data

c. Reservoir Size

Because of the odd shape of the reservoir (a large diameter at only one end and a quite small diameter at other end), the large diameter and overall length will be considered instead of the total volume. Assuming that a diameter and stroke combined change occurs when its capacity changes, the reduction in these dimensions for a change in capacity will be:

Letting

$$V = KD^2S$$

where

V = reservoir fluid capacity.

D = large reservoir diameter.

S = reservoir piston stroke.

K = proportionately constant.

Then a 50% reduction in capacity will result in

$$\frac{V}{2} = \frac{1}{2} KD^2S$$

$$= K \left(\frac{D^2}{\sqrt{2}}\right)\left(\frac{S}{\sqrt{2}}\right)$$

$$= K \left(\frac{D}{1.19}\right)^2 \left(\frac{S}{1.41}\right) .$$

Using standard reservoir design procedures and a straight-line relationship, the calculated variations in length and diameter can be shown by Figure 35.

From the above data, reservoir length and diameter as a function of tubing fluid weight can be expressed by

$$REL = 9 + 0.18 (TUFLW)$$

$$RED = 5 + 0.04 (TUFLW)$$

d. Reservoir Reliability

Reservoir reliability will be assumed constant because the same configuration will be used in each case. DC-8 reliability data indicate a failure rate for bootstrap-type reservoirs to be on the order of 16 failures per 10^6 operating hours. Therefore:

$$RER = 16$$

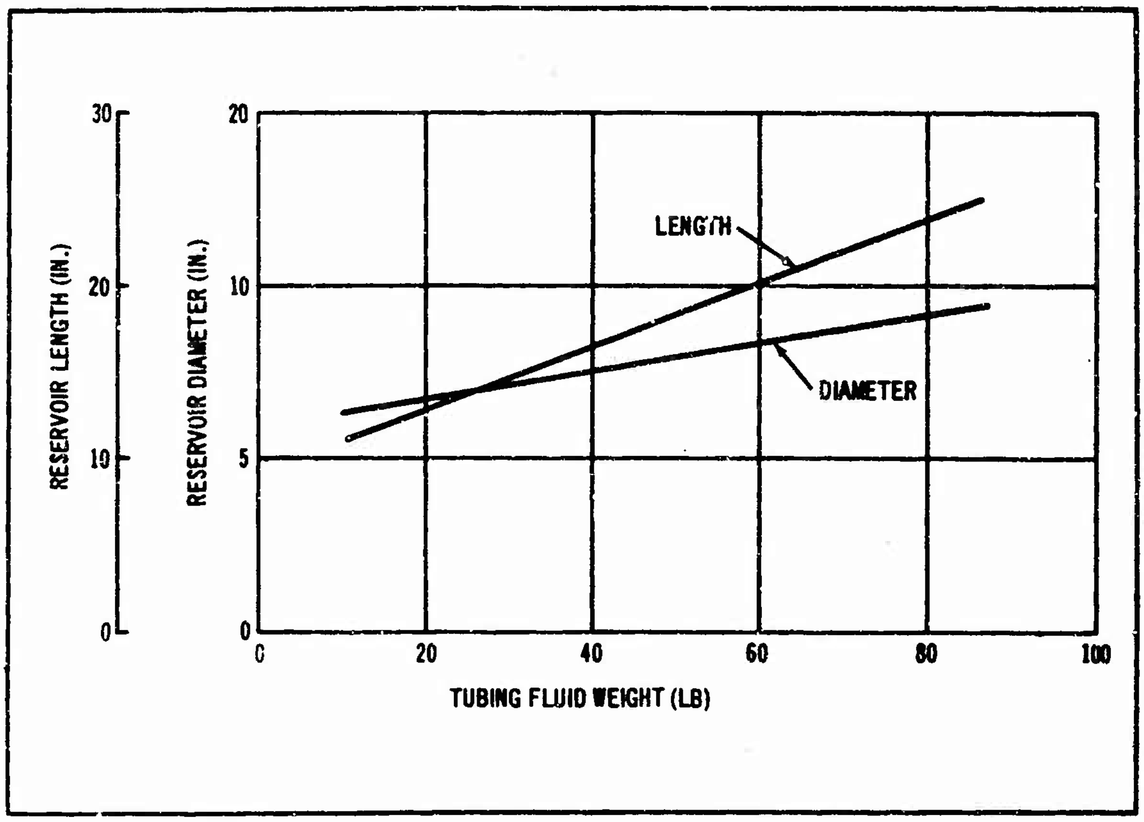


Figure 35. Reservoir Size Data

4. TUBING

Table IV summarizes the parameter relationships associated with the tubing. Derivations of these relationships follow the summary sheet.

a. Tubing Weight

Standard Douglas tube sizes used for a 3,000-psi system are shown in Table V.

The corresponding plumbing weight in lb/ft which includes fitting, clamps, and fluid can be approximated by the straight-line relationship shown in Figure 36.

TABLE IV

TUBING SUMMARY SHEET (Sheet 1 of 8)

PAGE ______

NAME: TUBING

SYMBOL T U

INPUTS:

T U W S 1	
T U W S 2	
T U W A L 1	
T U W A L 2	D P S 1
T U C S 1	D P S 2
T U C S 2	D P A L 1
T U C A L 1	D P A L 2
T U C A L 2	

OUPUTS

T U W

T U C

T U R

T U D P

T U D P 4

T U F L W

OUTPUT EQUATIONS

WEIGHT	T U W	=	TUWS1 + TUWS2 + TUWAL1 + TUWAL2
COST	T U C	=	TUCS1 + TUCS2 + TUCAL1 + TUCAL2
SIZE	N O N E		
RELIABILITY	T U R	=	44.
Pressure Loss	T U D P	=	DPS1 + DPS2 + DPAL1 + DPAL2
Pressure Loss at -40°F	T U D P 4	=	DPS14 + DPS24 + DPAL14 + DPAL24
Tubing Fluid Weight	T U F L W	=	TFWS1 + TFWS2 + TFWAL1 + TFWAL2

NOTES: Data obtained from Aircraft Division, Douglas Aircraft Co., Long Beach, California.

TABLE IV (Sheet 2 of 8)

PAGE ______

ADDITIONAL INPUTS

NAME: TUBING

SYMBOL T U

INPUTS:

D P S 1 4

D P S 2 4

D P A L 1 4

D P A L 2 4

T F W S 1

T F W S 2

T F W A L 1

T F W A L 2

OUPUTS

OUTPUT EQUATIONS

WEIGHT

COST

SIZE

RELIABILITY

NOTES:

TABLE IV (Sheet 3 of 8)

PAGE ______

NAME: #1 STEEL TUBING (PRESSURE SYSTEM)   SYMBOL: T U S 1

INPUTS:	OUTPUTS
D S 1 (≥.5)	T U W S 1
T U L S 1	T U C S 1
P R E S (>3000)	D P S 1
Q	D P S 1 A
	T F W S 1

OUTPUT EQUATIONS

WEIGHT	T U W S 1	= (.311 + 1.6*(DS1-.5))*TULS1*(1.+.00012*(PRES-3000))
COST	T U C S 1	= 1.3*DS1*TULS1*(1.+.00013*(PRES-3000.))
SIZE	N O N E	
RELIABILITY		See TU
Pressure Loss at -40°F	D P S 1 A	= 2.69*TULS1*Q*DS1**(-3.98)*(1.+.0002*(PRES-3000.))**2
Tubing Fluid Weight	T F W S 1	= .293*TULS1*((.430+.88*(DS1-.5))*(1.-2.4E-09*(PRES-3000.)**2)
Pressure Loss	D P S 1	= .001705*TULS1*(8.8*Q)**1.75*DS1**(-4.93)*(1.+.0003*(PRES-3000.)

NOTES: Equations for pressures greater than 3000 psi.

TABLE IV (Sheet 4 of 8)

PAGE ________

NAME: #1 STEEL TUBING (PRESSURE SYSTEM)

SYMBOL T U S 1

INPUTS:	OUPUTS
D S 1 (> .5)	T U W S 1
T U L S 1	T U C S 1
P R E S (< 3000)	D P S 1
Q	D P S 1 4
	T F W S 1

OUTPUT EQUATIONS

WEIGHT	T U W S 1	= (.311+1.6*(DS1-.5))*TULS1
COST	T U C S 1	= 1.3*DS1*TULS1
SIZE	N O N E	
RELIABILITY		See TU
Pressure Loss	D P S 1	= .001705*TULS1*(P.8*Q)**1.75*DS1**(-4.93)
Pressure Loss at -40°F	D P S 1 4	= 2.69*TULS1*Q*DS1**(-3.98)
Tubing Fluid Weight	T F W S 1	= .293*TULS1*(.430+.88*(DS1-.5))**2

NOTES: Equations for pressures equal to or less than 3000 psi.

TABLE IV (Sheet 5 of 8)

PAGE ______

NAME: #2 STEEL TUBING (PRESSURE SYSTEM)

SYMBOL: T U S 2

INPUTS: D S 2 (> .5); T U L S 2; P R E S (> 3000); Q

OUTPUTS: T U W S 2; T U C S 2; D P S 2; D P S 2 4; T F W S 2

OUTPUT EQUATIONS

WEIGHT	T U W S 2	= (.311+1.6*(DS2-.5))*TULS2*(1.+.00012*(PRES-3000.))
COST	T U C S 2	= 1.3*DS2*TULS2*(1.+.00013*(PRES-3000.))
SIZE	N O N E	
RELIABILITY		See TU
Pressure Loss at -40°F	D P S 2 4	= 2.69*TULS2*Q*DS2**(-3.98)*(1.+.0002*(PRES-3000.))**2
Tubing Fluid Weight	T F W S 2	= .293*TULS2*((.430+.88*(DS1-.5))*(1.-2.4E-09*(PRES-3000.)**2)
Pressure Loss	D P S 2	= .001705*TULS2*(4.4*Q)**1.75*DS2**(-4.93)*(1.+.0003*(PRES-3000.))

NOTES: Equations for pressures greater than 3000 psi.

TABLE IV (Sheet 6 of 8)

PAGE ______

NAME: #2 STEEL TUBING (PRESSURE SYSTEM)

SYMBOL: T U S 2

INPUTS: D S 2 (≥ .5); T U L S 2; P R E S (≤ 3000)

OUTPUTS: T U W S 2; T U C S 2; D P S 2; D P S 2 4; T F W S 2

OUTPUT EQUATIONS

WEIGHT	T U W S 2	= (.311+1.6*(DS2-.5))*TULS2
COST	T U C S 2	= 1.3*DS2*TULS2
SIZE	N O N E	
RELIABILITY		See TU
Pressure Loss	D P S 1	= .001705*TULS2*(4.4*Q)**1.75*DS2**(-4.93)
Pressure Loss at -40°F	D P S 1 4	= 1.35*TULS2*Q*DS2**(-3.98)
Tubing Fluid Weight	T F W S 2	= .293*TULS2*(.430+.88*(DS2-.5))**2

NOTES: Equations for pressures equal to or less than 3000 psi.

TABLE IV (Sheet 7 of 8)

PAGE ______

NAME: #1 ALUMINUM TUBING (RETURN SYSTEM)

SYMBOL: T U A L 1

INPUTS:	OUTPUTS
D A L 1 (≥ .5)	T U W A L 1
T U L A L 1	T U C A L 1
Q	D P A L 1
	D P A L 1 4
	T F W A L 1

OUTPUT EQUATIONS

WEIGHT	T U W A L 1	=	(.162+.695*(DAL1-.5))*TULAL1
COST	T U C A L 1	=	.222*DAL1*TULAL1
SIZE	N O N E		
RELIABILITY			See TU
Pressure Loss	D P A L 1	=	.001705*TULAL1* (8.8*Q)**1.75*DAL1**(-4.93)
Pressure Loss at -40°F	D P A L 1 4	=	2.69*TULAL1*Q*DAL1**(-3.98)
Tubing Fluid Weight	T F W A L 1	=	.293*TULAL1*(.430+.88*(DB1-.5))**2

NOTES:

TABLE IV (Sheet 8 of 8)

PAGE ________

NAME: #2 ALUMINUM TUBING (RETURN SYSTEM)

SYMBOL: T U A L 2

INPUTS: D A L 2 (≥.5)
T U L A L 2
Q

OUTPUTS: T U W A L 2
T U C A L 2
D P A L 2
D P A L 2 4
T F W A L 2

OUTPUT EQUATIONS

WEIGHT	T U W A L 2	=	(.162+.695*(DAL2-.5))*TULAL2
COST	T U C A L 2	=	.222*DAL2*TULAL2
SIZE	N O N E		
RELIABILITY			See TU
Pressure Loss	D P A L 2	=	.001705*TULAL2*(4.4*Q)**1.75*DAL2**(-4.93)
Pressure Loss at -40°F	D P A L 2 4	=	1.35*TULAL2*Q*DAL2**(-3.98)
Tubing Fluid Weight	T F W A L 2	=	.293*TULAL2*Q*(.430+.80*(D82-.5))**2

NOTES:

Table V
STANDARD PLUMBING SIZES

Tube size (in.)	Wall thickness Steel (supply system)	Aluminum (return system)
1/2	0.035	0.042
5/8	0.042	0.049
3/4	0.049	0.049
1	0.065	0.049

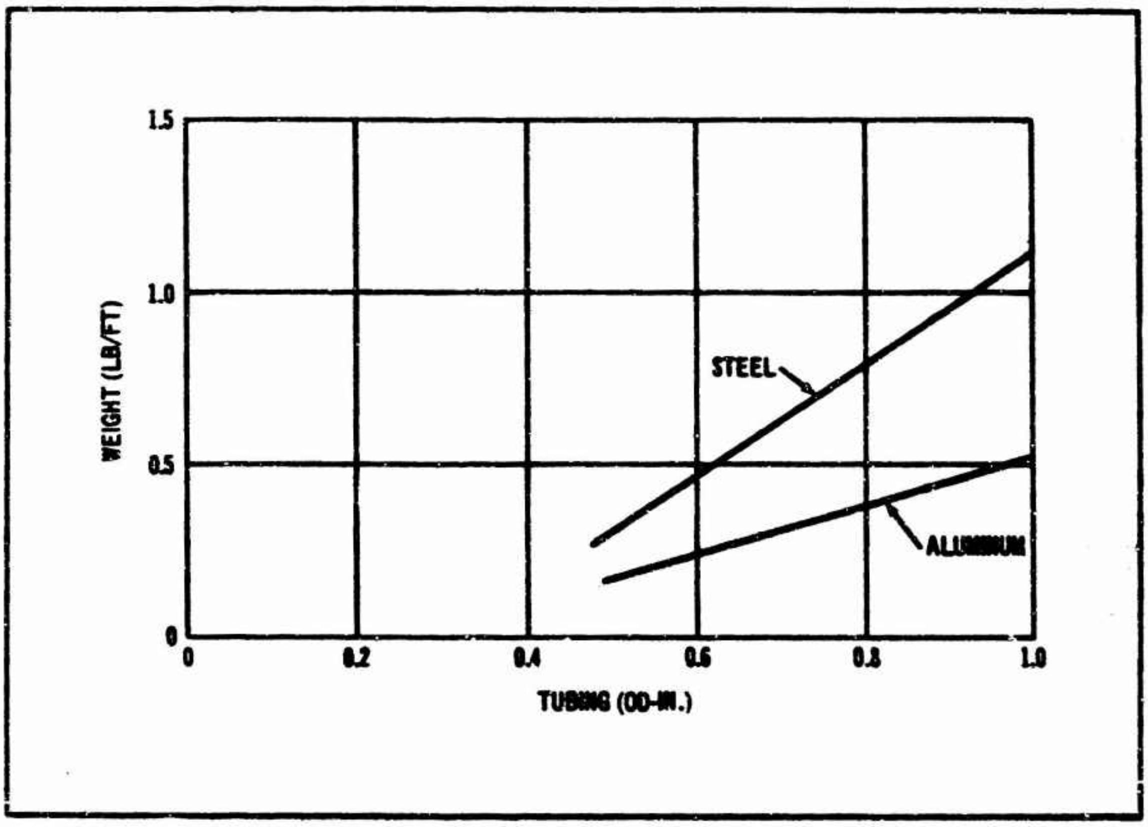


Figure 36. Plumbing Weight Data

Based on the above data, plumbing weight in lb/ft of length for a 3,000-psi system as a function of the tubing's outside diameter can be expressed as follows:

For steel

$$\text{TUWSF} = 0.311 + 1.6\,(\text{DS} - 0.5)$$

For aluminum

$$\text{TUWALF} = 0.162 + 0.695\,(\text{DAL} - 0.5)$$

where

$$\text{DS and DAL} \geq 0.5$$

As the plumbing is divided into two categories, the total plumbing weight will be:

$$\text{Total tubing weight} = \text{No. 1 steel} + \text{No. 2 steel} + \text{No. 1 aluminum} + \text{No. 2 aluminum}$$

$$\text{TUW} = \text{TUWS 1} + \text{TUWS 2} + \text{TUWAL 1} + \text{TUWAL 2}$$

where

$$\text{TUWS 1} = [0.311 + 1.6\,(\text{DS 1} - 0.5)]\ \text{TULS 1}.$$

$$\text{TUWS 2} = [0.311 + 0.16\,(\text{DS 2} - 0.5)]\ \text{TULS 2}.$$

$$\text{TUWAL 1} = [0.162 + 0.695\,(\text{DAL 1} - 0.5)]\ \text{TULAL 1}.$$

$$\text{TUWAL 2} = [0.162 + 0.695\,(\text{DAL 2} - 0.5)]\ \text{TULAL 2}.$$

To maintain about the same stress levels in the tubing at various system pressures, wall thicknesses will change, and this, in turn, will affect the weight. Currently, all the required combinations of standard wall thicknesses and tube diameters for the higher pressure systems are not available. However, it will be assumed that they would become available if the need existed in quantities to make their manufacture worthwhile. The diameter and wall thickness combinations which could be used for the steel pressure lines are shown in Table VI.

Table VI

PLUMBING SIZES AS A FUNCTION OF SYSTEM PRESSURE

Tube size (in.)	Pressures (psi) 3,000	4,000	5,000	6,000
	Wall thickness			
1/2	0.035	0.049	0.058	0.065
5/8	0.042	0.058	0.065	0.083
3/4	0.049	0.065	0.083	0.095
1	0.065	0.083	0.095	0.120

A weight calculation indicates an increase of about 12% at all tube sizes for each 1,000-psi increase in pressure. As a result, the plumbing weight expressed as a function of tubing diameter and system pressure is

$$\text{TUW} = [\text{TUWS } 1 + \text{TUWS } 2]\,[1 + 0.00012\,(\text{PRES} - 3{,}000)] + \text{TUWAL } 1 + \text{TUWAL } 2$$

where

$$(\text{PRES} - 3{,}000) \geq 0 \ .$$

Because the return pressures will not appreciably change regardless of the supply pressure, it will be assumed that the return system plumbing weight will not change as a function of pressure.

Because the reservoir design is a function of the fluid weight in the tubing, the weight of the fluid will be calculated separately.

Using the following expression between the tubing inside and outside diameter for a 3,000-psi system.

$$\text{TUID} = 0.430 + 0.88\,(D - 0.5)$$

where

$$D \geq 0.5 \ ,$$

the fluid weight per foot of length expressed as a function of the tubing outside diameter is

$$TUFLWF = 0.293\,[0.430 + 0.88\,(D - 0.5)]^2$$

Fluid specific gravity was assumed constant at a value of 0.86. It was also assumed that the effect of the difference of wall thickness between the steel and aluminum tubing for a 3,000-psi system is negligible, so that the above expression would apply to both types of tubing.

Separating each tube category results in the following expressions:

$$TUFLW = TFWS\ 1 + TFWS\ 2 + TFWAL\ 1 + TFWAL\ 2$$

where

$$TFWS\ 1 = 0.293\,(TULS\ 1)\,[0.430 + 0.88\,(DS\ 1 - 0.5)]^2$$

$$TFWS\ 2 = 0.293\,(TULS\ 2)\,[0.430 + 0.88\,(DS\ 2 - 0.5)]^2$$

$$TFWAL\ 1 = 0.293\,(TULAL\ 1)\,[0.430 + 0.88\,(DAL\ 1 - 0.5)]^2$$

$$TFWAL\ 2 = 0.293\,(TULAL\ 2)\,[0.430 + 0.88\,(DAL\ 2 - 0.5)]^2$$

Use of the higher-pressure systems with the thicker-walled tubing results in less fluid weight. Referring to Table VI, the reduction in inside diameter for each 1,000-psi increase in pressure regardless of tube size, averages out to be 4.9%. As a result, (TUFLW) with a pressure term added will be

$$TUFLW = [TFWS\ 1 + TFWS\ 2]\,[1 - 2.4 \times 10^{-9}\,(PRES - 3,000)^2] + TFWAL\ 1 + TFWAL\ 2$$

where

$$(PRES - 3,000) \geq 0\ .$$

b. Tubing Cost

Costs for tubing, fittings, clamps, and an arbitrary quantity can be approximated by the straight-line relationship as shown in Figure 37.

Based on the foregoing data, plumbing cost in dollars per foot of length for a 3,000-psi system as a function of the tubing's outside diameter can be expressed as follows:

For steel

$$TUCSF = 1.3\,(DS)$$

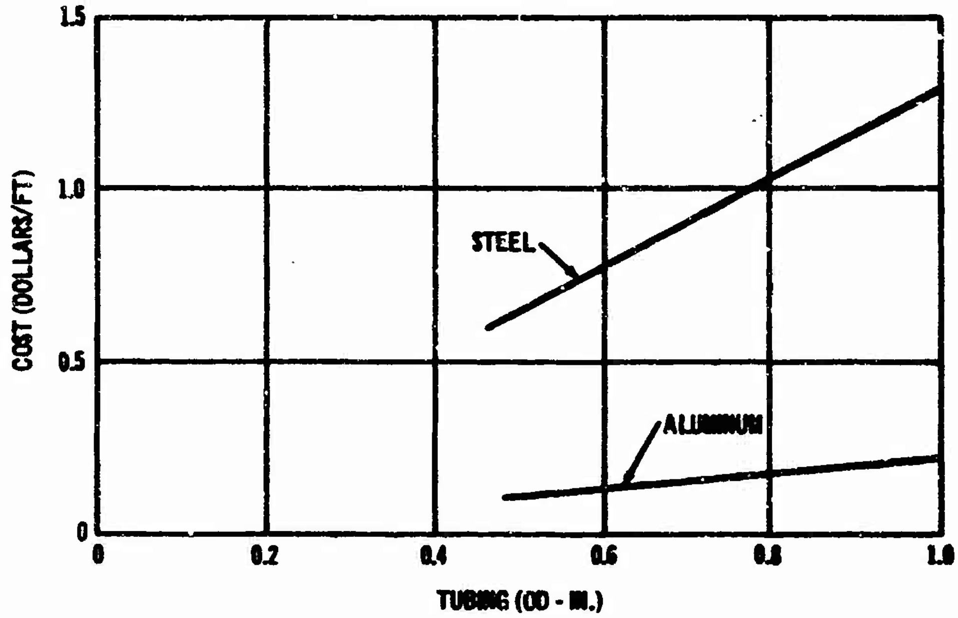


Figure 37. Plumbing Cost Data

For aluminum

$$TUCALF = 0.222 \text{ (DAL)}$$

As a result of the division of plumbing lengths, the total cost will be

$$TUC = TUCS\ 1 + TUCS\ 2 + TUCAL\ 1 + TUCAL\ 2$$

where

$$TUCS\ 1 = 1.3 \text{ (DS 1) (TULS 1)}$$

$$TUCS\ 2 = 1.3 \text{ (DS 2) (TULS 2)}$$

$$TUCAL\ 1 = 0.222 \text{ (DAL 1) (TULAL 1)}$$

$$TUCAL\ 2 = 0.222 \text{ (DAL 2) (TULAL 2)}$$

Calculations indicate about a 13% increase in cost for each 1,000-psi increase in pressure above 3,000 psi. As a result, the plumbing cost expressed as a function of tubing diameter and system pressure is

$$\mathrm{TUC} = [\mathrm{TUCS\ 1} + \mathrm{TUCS\ 2}]\,[1 + 0.00013\,(\mathrm{PRES} - 3,000)] + \mathrm{TUCAL\ 1} + \mathrm{TUCAL\ 2}$$

where

$$(\mathrm{PRES} - 3,000) \geq 0 \ .$$

c. Tubing Size

Tubing volume (and the space it occupies) normally does not affect the corresponding vehicle envelope requirements for this type of aircraft; therefore, these factors will not be considered in this study.

d. Tubing Reliability

Tubing reliability will be considered constant regardless of tube size because the thicker wall tubing is used with the higher pressures. Thus, for each case, about the same maximum stress level is retained. A failure rate obtained from a FARADA Handbook for a horizontal stabilizer hydraulic tubing assembly is 44 failures per 10^6 operating hours. Therefore,

$$\mathrm{TUR} = 44$$

e. Tubing Pressures Loss

The tubing pressure losses and fluid temperatures are interrelated. Therefore, to calculate these losses, a fluid temperature will be assumed. Because this is a simplified hydraulic system and to reduce the complexity of the pressure loss calculations, the fluid temperature will be assumed to be a constant. Based upon the hydraulic system design capabilities generated during the C-5A proposal effort, a fluid temperature of 120°F will be used.

With 120°F fluid and flows in the low-turbulent region, the pressure loss can be approximated by a straight-line relationship on a log-log plot, as shown by Figure 38.

Based on these data, tubing pressure loss per foot of length as a function of tubing flow and tubing outside diameter for a 3,000-psi system can be expressed by

$$\mathrm{TUDPF} = (\mathrm{TUQ})^{1.75}\left[0.00155\,(D)^{-4.93}\right]$$

It was assumed that the effect of the difference in wall thickness between the steel and aluminum tubing for a 3,000-psi system is negligible, so that the above expression would apply to both types of tubing.

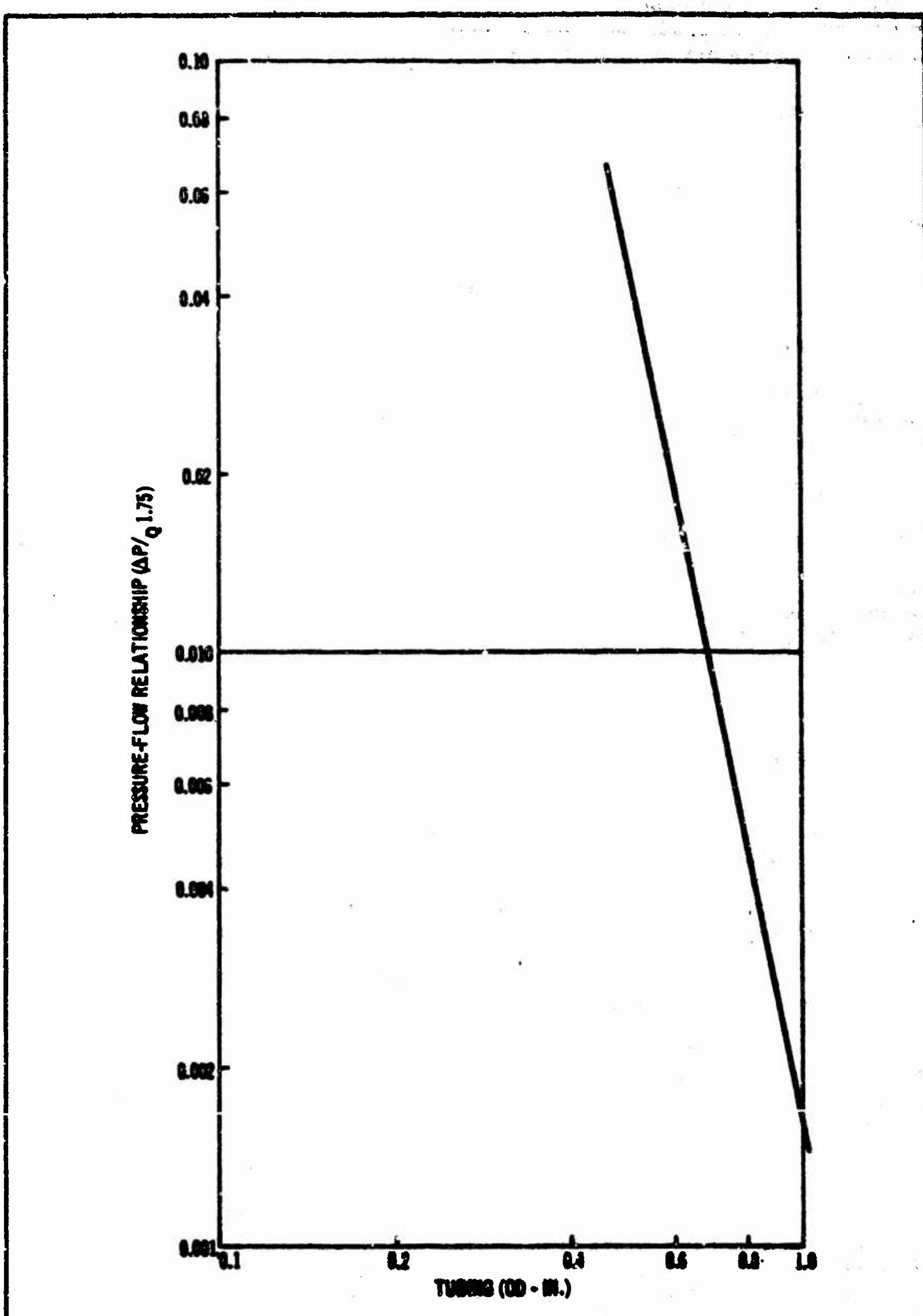


Figure 38. Tubing Pressure Loss Data/Foot

Adding 10% of (TUDPF) for pressure losses in plumbing bends, fittings, and other small components not considered in this study, and separating each tube category, results in the following expressions for line loss as a function of valve flow, line length, and line diameter.

$$\text{TUDP} = \text{DPS 1} + \text{DPS 2} + \text{DPAL 1} + \text{DPAL 2}$$

where

$$\text{DPS 1} = 0.001705\ (\text{TULS 1})\ (8.8\text{Q})^{1.75}\ (\text{DS 1})^{-4.93}$$

$$\text{DPS 2} = 0.001705\ (\text{TULS 2})\ (4.4\text{Q})^{1.75}\ (\text{DS 2})^{-4.93}$$

$$\text{DPAL 1} = 0.001705\ (\text{TULAL 1})\ (8.8\text{Q})^{1.75}\ (\text{DAL 1})^{-4.93}$$

$$\text{DPAL 2} = 0.001705\ (\text{TULAL 2})\ (4.4\text{Q})^{1.75}\ (\text{DAL 2})^{-4.93}$$

Use of the higher-pressure systems with the thicker-walled tubing results in smaller flow areas and more pressure loss. Referring to Table III, the reduction in inside diameter for each 1,000-psi increase in pressure regardless of tube size averages out to be 4.9%. The corresponding increase in pressure loss is about 30%. As a result, (TUDP) with a pressure term added will be:

$$\text{TUDP} = [\text{DPS 1} + \text{DPS 2}]\ [1 + 0.0003\ (\text{PRES} - 3{,}000)] + \text{DPAL 1} + \text{DPAL 2}$$

where

$$(\text{PRES} - 3{,}000) \geq 0\ .$$

f. Tubing Size Limitation

Because a relatively high fluid temperature was used for the tubing pressure loss calculations, selection of a low limit on tube size was necessary to ensure that the system would have sufficient actuator ΔP to perform satisfactorily at lower temperatures. For this application, a low temperature of -40°F has been selected, and satisfactory performance at -40°F has been defined as being able to obtain about 25% of maximum control surface rate with zero hinge moment. This means that the tubing pressure loss at -40°F cannot exceed the difference between the supply pressure and the required valve ΔP. In an actual application, the actuation system performance should be checked at various fluid temperatures after the tubing sizes have been selected to ensure that the system meets the different satisfactory performance requirements whatever they may be.

Using a laminar flow viscosity correction factor for a nominal pressure of 3,000 psi, pressure loss data can be approximated by the straight-line relationship plotted on log-log paper and shown by Figure 39. Based on these data, the pressure loss at $-40^{\circ}F/ft$ of length for a 3,000-psi system as a function of tubing flow and outside diameter can be expressed by:

$$\text{TUDPF} = (\text{TUQ})\ (0.306)\ (\text{D})^{-3.98}$$

It was assumed that the effect of the difference of wall thickness between the steel and aluminum tubing for a 3,000-psi system is negligible so that the above expression would apply to both types of tubing.

Adding 10% of (TUDPF) for pressure losses in plumbing bends, fittings, and other small components not considered in this study and separating each tube category results in the following expressions for line loss as a function of valve flow, line length, and line diameter.

$$\text{TUDP4} = \text{DPS14} + \text{DPS24} + \text{DPAL14} + \text{DPAL24}$$

where

$$\text{DPS14} = 2.69\ (\text{TULS1})\ (\text{Q})\ (\text{DS1})^{-3.98}$$

$$\text{DPS24} = 1.35\ (\text{TULS2})\ (\text{Q})\ (\text{DS2})^{-3.98}$$

$$\text{DPAL14} = 2.69\ (\text{TULAL1})\ (\text{Q})\ (\text{DAL1})^{-3.98}$$

$$\text{DPAL24} = 1.35\ (\text{TULAL2})\ (\text{Q})\ (\text{DAL2})^{-3.98}$$

Use of the higher-pressure systems with the thicker-walled tubing results in more pressure losses because of smaller flow areas and a larger viscosity correction factor. Referring to Table VI, the reduction in the inside diameter for each 1,000-psi increase in pressure regardless of tube size, averages out to be 4.9%. The corresponding increase to pressure loss is about 20% because of the tubing and also 20% because of the viscosity correction. As a result, (TUDP4) with a pressure term added will be

$$\text{TUDP4} = [\text{DPS14} + \text{DPS24}]\ [1 + 0.0002\ (\text{PRES} - 3{,}000)]^{2}$$
$$+ \text{DPAL14} + \text{DPAL24}$$

where

$$(\text{PRES} - 3{,}000) \geq 0$$

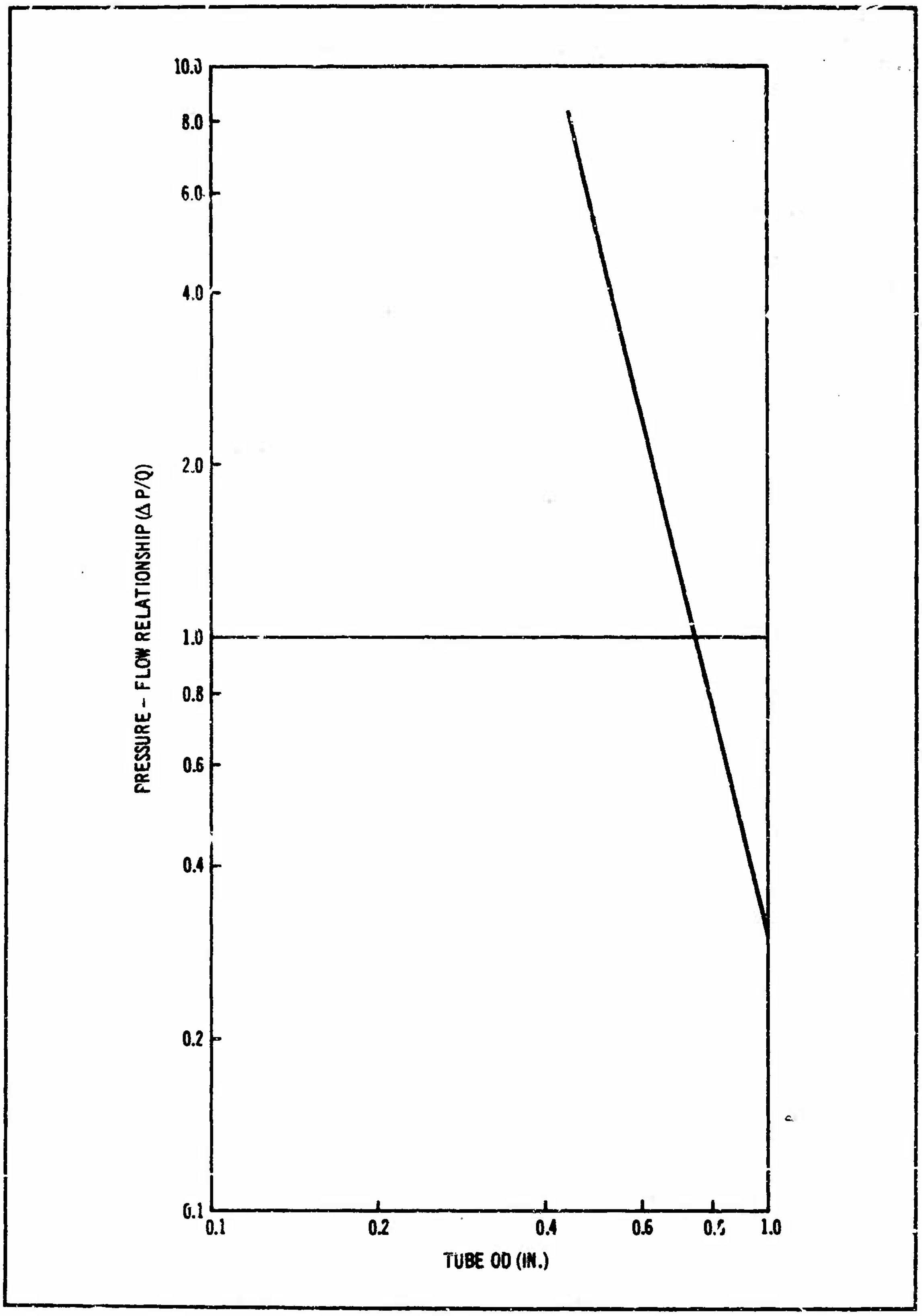


Figure 39. Tubing Pressure Loss Data/Foot at a Fluid Temperature of –40°F

5. ACTUATOR

Table VII summarizes the parameter relationships associated with the actuator. Derivations of these relationships follow the summary sheet.

a. Actuator Weight

A schematic of the tandem actuator unit is shown by Figure 40. Sample weight calculations based upon this schematic and plotted as a function of torque arm length with the supply pressure as a parameter indicate a relationship as shown by Figure 41. Application of a hyperbolic relationship to these curves with a minimum weight correction factor results in the following expression:

$$ACTW = HMT\,(10^{-5})\left[\left(0.0625\left[1 + 4\left(RNOM - 56[PNOM]^{-0.280}\right)^2\right]\right)^{0.5} + 140\,(PNOM)^{-0.242}\right]$$

To obtain the above expression, the ratio of the actuator ΔP (ACTDP) to the supply pressure (PRES) was approximated by a value of two-thirds. This ratio was used because it approximates the pressure where maximum power transfer occurs. Because the system flow-pressure relationships are interrelated, an actuator ΔP approximation was necessary to calculate the actuator area.

In the above expression, actuator design is based upon a constant torque design. During the optimization process, however, the actuator weight for variable torque design will have to be determined because arbitrarily selected values of supply pressure, actuator area, and torque arm length will be used. Corresponding torque correction factors must be added to this expression. With use of the same type of calculations, the actuator area factor can be approximated by

$$+3.3\,(AREA-ANOM)\,(10^{-5})$$

The supply pressure factor can be approximated by

$$+0.95\,(PRES-PNOM)\,(10^{-8})$$

The torque arm factor can be approximated by

$$+1.2\,(R-RNOM)\,(10^{-5})$$

TABLE VII
ACTUATOR SUMMARY SHEET

PAGE ______

NAME: ACTUATOR

SYMBOL A C T

INPUTS: RH M

A R E A

P R E S

D E L M X

R N O M

P N O M

A N O M

OUTPUTS A C T W

A C T L

OUTPUT EQUATIONS

WEIGHT	A C T W	=	HM*.00002*((0625*(1.+4.*(RNOM-56.*(PNOM)**(-.280))**2.))**.5+140.*PNOM**(-.242)+3.3*(AREA-ANOM) + .00095*(PRES-PNOM)+1.2*(R-RNOM))
COST			See VAP
SIZE			
RELIABILITY			See VAP
*LENGTH	A C T L	=	.0524*DELMX*R+13.

NOTES: Data obtained from Bertea Corporation, Irvine, California, and from Aircraft Division, Douglas Aircraft Company, Long Beach, California.

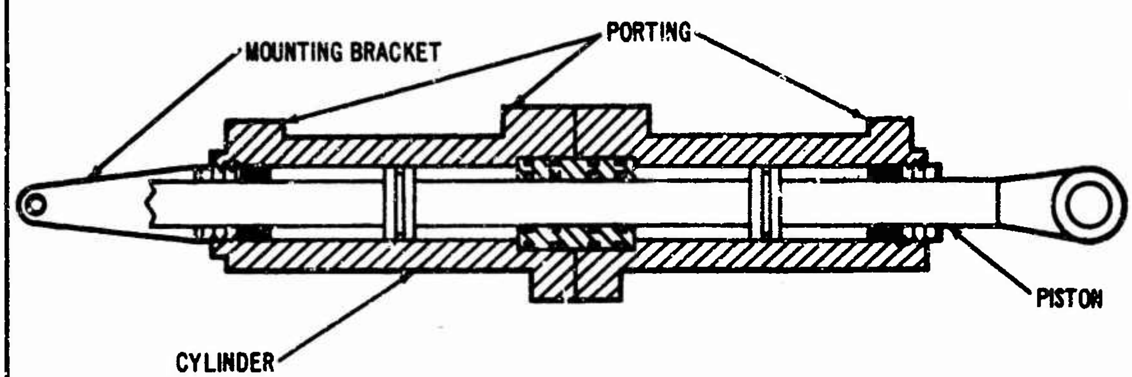


Figure 40. Actuator Schematic

Incorporation of the (HM) notation

$$HM = (1/2)\ HMT$$

results in the final expression

$$\begin{aligned} ACTW = 2\ HM(10^{-5}) \Bigg[& \left(0.0625 \left[1 + 4\left(RNOM - 56[PNOM]^{0.280}\right)^2\right]\right)^{0.5} \\ & + 140\ (PNOM)^{-0.242} + 3.3\ (AREA-ANOM) \\ & + 0.95\ (PRES-PNOM)\ (10^{-3}) + 1.2\ (R-RNOM) \Bigg] \end{aligned}$$

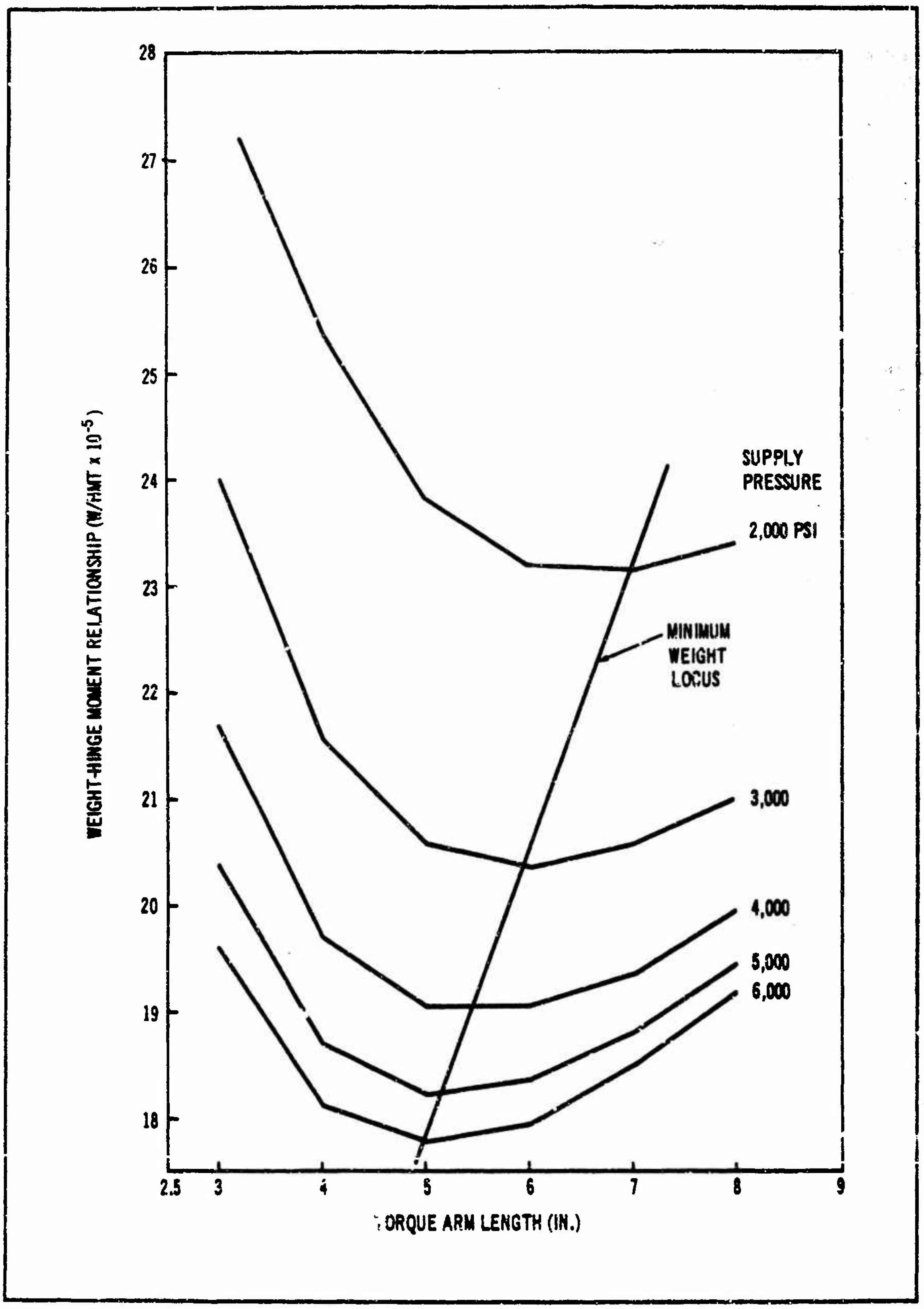


Figure 41. Actuator Weight Data

b. Actuator Cost

The actuator is an integral part of the valve actuator package so its cost will be included in the total assembly.

c. Actuator Size

Because the valve assembly more or less surrounds the actuator, the actuator's area is not an important dimension. The length is predominant, however, and will be considered. The actuator length is primarily a function of the stroke. Other factors, such as method of attachment, cylinder ends, piston thickness, etc., influence actuator length; however, for a particular configuration they will be fairly constant. As a result, actuator length will be defined in terms of a constant plus a stroke function.

Assuming a simple small angle relationship, actuator stroke as a function of control surface deflection can be expressed by

$$ACTS = DELMX\ (R)\ /57.3$$

Because the actuator is a tandem unit, the piston rod must be at least three times as long as the calculated stroke (two cylinders + one stroke). For the Boeing 727 elevator-control package, it has a length factor of about 13 in. over and above that contributed by the stroke. As a result, the overall actuator length can be represented by:

$$ACTL = 0.0524\ (DELMX)\ (R) + 13$$

d. Actuator Reliability

As with the other components, published reliability data are very general, and so it is impossible to use it to obtain a correlation between failure rates and actuator design parameters. To determine whether actual failure reports would provide more useful information, it was decided to survey the ATA (Air Transport Association) failure summary reports and determine whether a correlation exists, for example, betw n actuator leakage failures and actuator stroke because stroke i a parameter being used in this study. ATA reports covering the period of 1 January 1961 through 31 December 1964 for the DC-8 totaling 1,271,201 hours of aircraft operation were reviewed. The results are tabulated in Table VIII.

Table VIII

DC-8 ACTUATOR LEAKAGE FAILURE DATA

Function	No. per plane	Nominal strokes	No. of failures
Main landing gear uplatch	2	0.6	3
Aileron	2	4.1	0
Tab lockout	2	4.2	1
Flap outboard	2	4.3	0
Flap midwing	2	5.2	0
Flap inboard	2	7.1	2
Main landing gear trim	2	7.4	1
Nose wheel steering	2	7.9	1
Rudder	1	7.9	0

It is not evident from the above data that there is a correlation between leakage failures and stroke if one does exist. A factor that complicates the situation is the incorporation of design changes during the 4-year period because it was impossible to determine the change effectively from the available data. It was noted during the survey, however, that when a failure was not diagnosed as an isolated incident and a design change occurred, this change usually corrected the situation and no more failures of this kind happened again.

As a result, because the actuator is an integral part of the valve actuator package, its reliability will be included in the reliability of the total assembly.

6. VALVE

Table IX summarizes the parameter relationships associated with the valve. Derivations of these relationships follow the summary sheet.

a. Valve Assembly Weight

The valve assembly contains a number of components. These include the following:

1. Transfer valve.

2. Autopilot actuator.

TABLE IX
VALVE SUMMARY SHEET

PAGE ______

NAME: VALVE ACTUATOR PACKAGE

SYMBOL V A P

INPUTS: A C T W
V A S W

OUPUTS V A P W
V A P C
V A P R

OUTPUT EQUATIONS

WEIGHT	V A P W	=	ACTW + VASW
COST	V A P C	=	7500.
SIZE			See ACT
RELIABILITY	V A P R	=	16.

NOTES: Data obtained from Bertea Corporation, Irvine, California.

3. Metering valve.

4. Tab lockout valve.

5. Filters.

6. Solenoids.

7. Bypass valves.

8. Input, feedback, and summing leakages.

9. Various manifolds.

Because this unit is similar to the Boeing 727 elevator-valve assembly, and, because it contains so many different functions, its weight will be expressed as a function of valve flow and based upon the weight of the Boeing 727 unit. The Boeing unit weighs about 35 lb and has a maximum valve flow of about 1 gpm, so, if 10% weight is added for each gpm above one, the valve assembly weight for a 3,000-psi system can be expressed by

$$VASW3 = 35 + (Q - 1)(0.10)(35)$$

For systems with pressures other than 3,000 psi, a nominal change in weight of 5% for each 1,000-psi change in system pressure will be used. Adding this pressure function to the previous expression results in

$$VASW = (31.5 + 3.5Q)[1 + 0.00005(PRES - 3,000)]$$

b. Valve Assembly Cost and Reliability

Because the valve assembly is an integral part of the valve actuator package, its cost and reliability will be included in the total unit.

c. Valve Assembly Size

The valve assembly size is quite flexible because its combination of valves, manifolds, and so forth, can be patched together in many different possible ways. As a result, it can be made to fit most envelopes and so the volume or size of the unit will not be considered.

7. VALVE ACTUATOR

Table X summarizes the parameter relationships associated with the valve actuator. Derivations of these relationships follow the summary sheet.

TABLE X

VALVE ACTUATOR SUMMARY SHEET

PAGE ______

NAME: VALVE ASSEMBLY

SYMBOL V A S

INPUTS: P R E S
Q

OUTPUTS V A S W

OUTPUT EQUATIONS

WEIGHT	V A S W	= (31.5 + 1.5*Q)*(1. + .000054(PRES-3000.))
COST		See VAP
SIZE	N O N E	
RELIABILITY		See VAP

NOTES: Data obtained from Bertea Corporation, Irvine, California.

a. Valve Actuator Package Weight

This unit is a combination of the valve assembly and actuator and so its weight can be expressed by

$$VAPW = ACTW + VASW$$

b. Valve Actuator Package Cost and Reliability

Cost and reliability will remain about the same for the various systems that are studied because the same configuration will be used in each case. Data obtained from Bertea Products using the Boeing 727 unit as a guide indicate that for an arbitrary quantity, the cost in dollars is

$$VAPC = 7,500$$

The un . reliability based upon data obtained from various commercial airlines for the Boeing 727 unit over a period of 20 months and a total service time of 188,441 flight hours indicates the number of failures per 10^6 hours to be

$$VAPR = 16$$

c. Valve Actuator Package Size

The predominant size factor in this assembly is the actuator length. As mentioned for the valve assembly, the overall size is not considered because the design is very flexible and can usually be made to fit most envelopes.

d. Valve Actuator Package Pressure Distribution

System pressure can be divided into the following three parts: (1) plumbing ΔP, (2) actuator ΔP, and (3) valve ΔP, as shown by Figure 31. The flow-actuator rate relationship is a function of the actuator area which, in turn, is a function of the actuator ΔP. As shown by Figure 42, however, the available actuator ΔP depends upon the flow. As a result, an assumption must be made, values calculated, the assumption compared to the calculated value, and the process repeated if the comparison reveals too large an error. A method that can be used to obtain a good first assumption is as follows:

At maximum rate

$$PRES = TUDPM + VASDPM$$

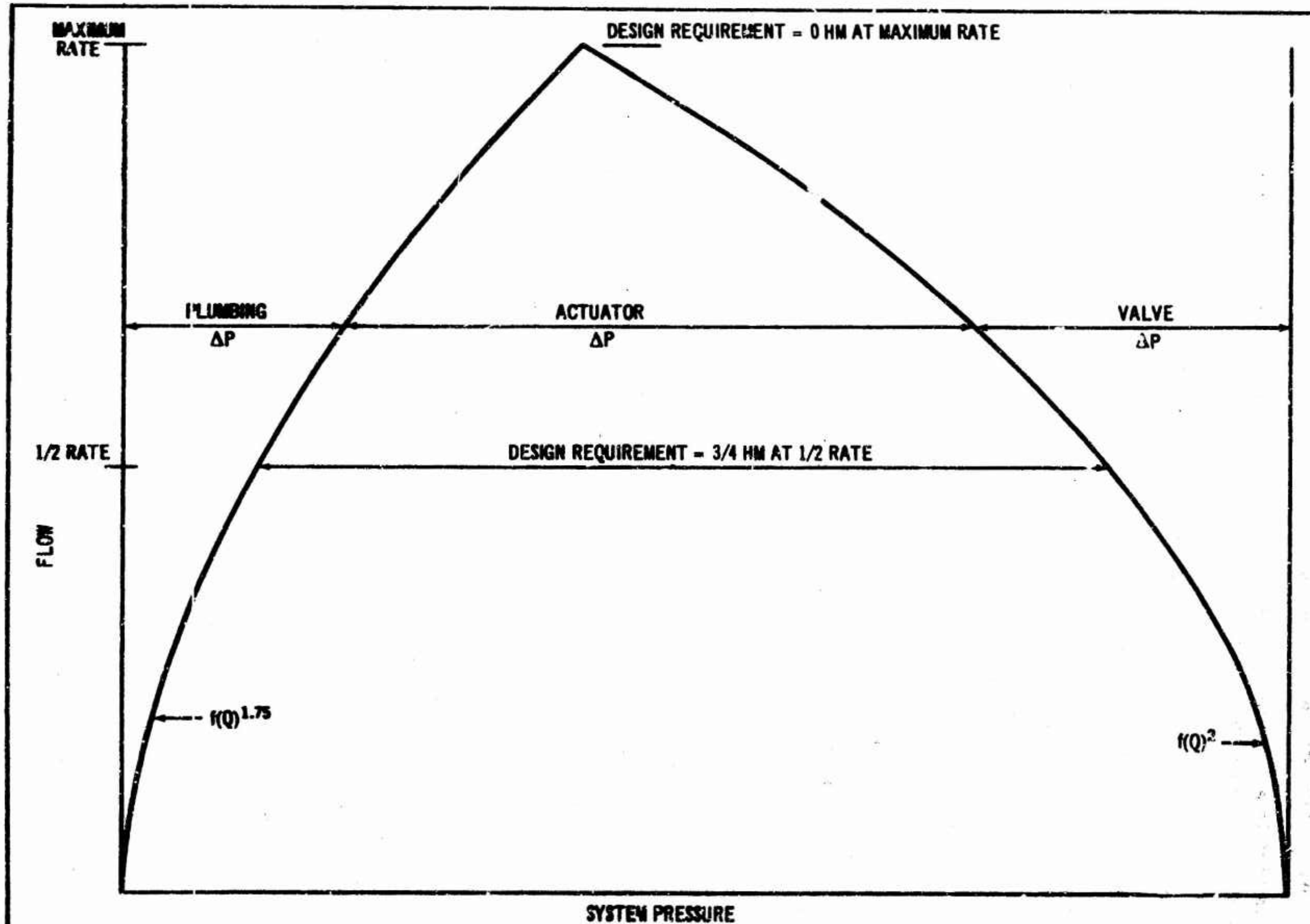


Figure 42. System Pressure Distribution

because there isn't any actuator ΔP. If it is assumed that the plumbing ΔP is a function of Q^2 instead of $Q^{1.75}$, then at one-half rate the actuator ΔP can be defined as

$$\begin{aligned} \text{ACTDP1} &= \text{PRES} - (1/4)\ \text{TUDPM} - (1/4)\ \text{VASDPM} \\ &= \text{PRES} - 1/4\ [\text{TUDPM} + \text{VASDPM}] \\ &= 3/4\ \text{PRES} \end{aligned}$$

With a value for actuator ΔP, actuator area at the one-half rate design condition can be expressed by

$$\text{AREA} = \text{HMD/ACTDP1}\ (\text{R})$$

The corresponding valve flow in gallons per minute can be expressed by

$$Q = 0.00453\ (\text{DEFLD})\ (\text{AREA})\ (\text{R})$$

From this information, a new value for the actuator ΔP (ACTDP) at the one-half-rate design condition can be calculated

$$\text{ACTDP} = \text{PRES} - 0.25\ (\text{PRES} - \text{TUDPM}) - \text{TUDP2}$$

To determine whether an iteration is required, the following ground rule was established:

if (ACTDP1)-(ACTDP) $\leq$ 100, then use ACTDP as calculated,
if (ACTDP1)-(ACTDP) > 100, then substitute (ACTDP1 + ACTDP)/2 for the old value of ACTDP1 and recalculate ACTDP.

8. FLUID TEMPERATURE

A heat exchanger was not included in this simplified actuation system because a cursory fluid temperature check indicated that the fluid temperature would not exceed the nominal specified limit of 160°F. The following calculations are a sample of the type used to check the fluid temperature. When used on a DC-8, temperatures obtained utilizing these calculations compared favorably with actual test data. The following assumptions will be used:

1. Overall pump efficiency and all system pressure losses with the exception of actuator ΔP go into heating the fluid.

2. Nominal flow of one-third maximum pump flow.

3. Actuator ΔP of three-fourths system pressure.

4. All of the heat lost will be by convection with a surface coefficient (C) of 2 Btu/hr ft^2 °F.

5. Surface area will consist of tubing area plus 10% for other components.

6. A standard sea level ambient day temperature (T) of 68°F will be used because ground operation will give the higher fluid temperatures.

With a hinge moment of 36,000 in.-lb, a control surface rate of 40°/sec, and a system pressure of 3,000 psi, the nominal system flow (QN) will be

$$QN = \frac{(0.00453)(DELD)(HMD)(1/3)(8.8)}{(3/4)(PRES)}$$

$$= \frac{(0.00453)(40)(36,000)(1/3)(8.8)}{(3/4)(3,000)}$$

$$= 8.5 \text{ gpm}$$

The hydraulic system output power (HPO) based upon the actuator ΔP will be

$$HPO = \frac{(3/4)(PRES)(QN)}{1714}$$

$$= \frac{(3/4)(3,000)(8.5)}{1714}$$

$$= 11.15 \text{ hp}$$

The input power (HPI) with an overall pump efficiency of 92% will be

$$HPI = \frac{(PRES)(QN)}{1714\ (0.92)}$$

$$= \frac{(3,000)(8.5)}{1714\ (0.92)}$$

$$= 16.15 \text{ hp}$$

Therefore, the heat generated (HIN) will be

$$HIN = (HPI\text{-}HPO)\ 2547\ \frac{\text{Btu/hr}}{\text{hp}}$$

$$= (16.15 - 11.15)(2547)$$

$$= 12,730 \text{ Btu/hr.}$$

The tubing surface area (TSA) will be calculated on the basis of using a 5/8-in.-diam for the No. 1 tubing and 1/2-in.-diam for the No. 2 tubing. As a result, the total surface area (SA) will be

$$SA = 1.1\ (TSA)$$

$$= \frac{(1.1)(\pi)}{12}\ [(0.625)\ (370) + (0.5)\ (160)]$$

$$= 89.5\ ft^2$$

The fluid temperature obtained with 12,730 Btu/hr heat input, 89.5 ft^2 of surface area, and 68°F ambient will be

$$HIN = (C)\ (SA)\ (FLT - T)$$

$$12,730 = 2(89.5)\ (FLT - 68)$$

$$FLT = 139^{o}F$$

9. SYSTEM ELASTICITY

Table XI summarizes the parameter relationships associated with the system elasticity. Derivations of these relationships follow the summary sheet.

The actuation system resonant frequency is a function of the system elasticity, so relationships between it and the system design parameters must be obtained. In this study, those items which will be used to describe system elasticity are fluid elasticity and the elasticity of the actuator cylinders. They are like springs in series and the effective elasticity of the system can be expressed by

$$\frac{1}{SE} = \frac{1}{FLE} + \frac{1}{CYE}$$

10. FLUID ELASTICITY

Table XII summarizes the parameter relationships associated with the fluid elasticity. Derivations of these relationships follow the summary sheet.

The fluid-bulk modulus (or elasticity) is a function of the fluid temperature and pressure. Data obtained by R. L. Peeler and J. Green (Reference 5) on the adiabatic bulk modulus of MIL-H-5606A fluid, covering a temperature range of 50° to 200°F, can be approximated by Figure 43. Based on these data, the fluid bulk modulus as a function of the fluid temperature and system pressure can be expressed by

$$FLE = -610\ (FLT) + 138500\ [(2/3)(PRES)]^{0.1082}$$

TABLE XI

SYSTEM ELASTICITY SUMMARY SHEET

PAGE ________

NAME: SYSTEM ELASTICITY

SYMBOL S E

INPUTS: F L E

C Y E

OUPUTS S E

OUTPUT EQUATIONS

WEIGHT

COST

SIZE

RELIABILITY

ELASTICITY S E = FLE*CYE/(FLE + CYE)

NOTES:

TABLE XII

FLUID ELASTICITY SUMMARY SHEET

PAGE ______

NAME: FLUID ELASTICITY

SYMBOL F L E

INPUTS: P R E S F L T

OUTPUTS F L E

OUTPUT EQUATIONS

WEIGHT

COST

SIZE

RELIABILITY

ELASTICITY F L E = -610.*FLT + 138500.* (2.713.*PRES)**.1089

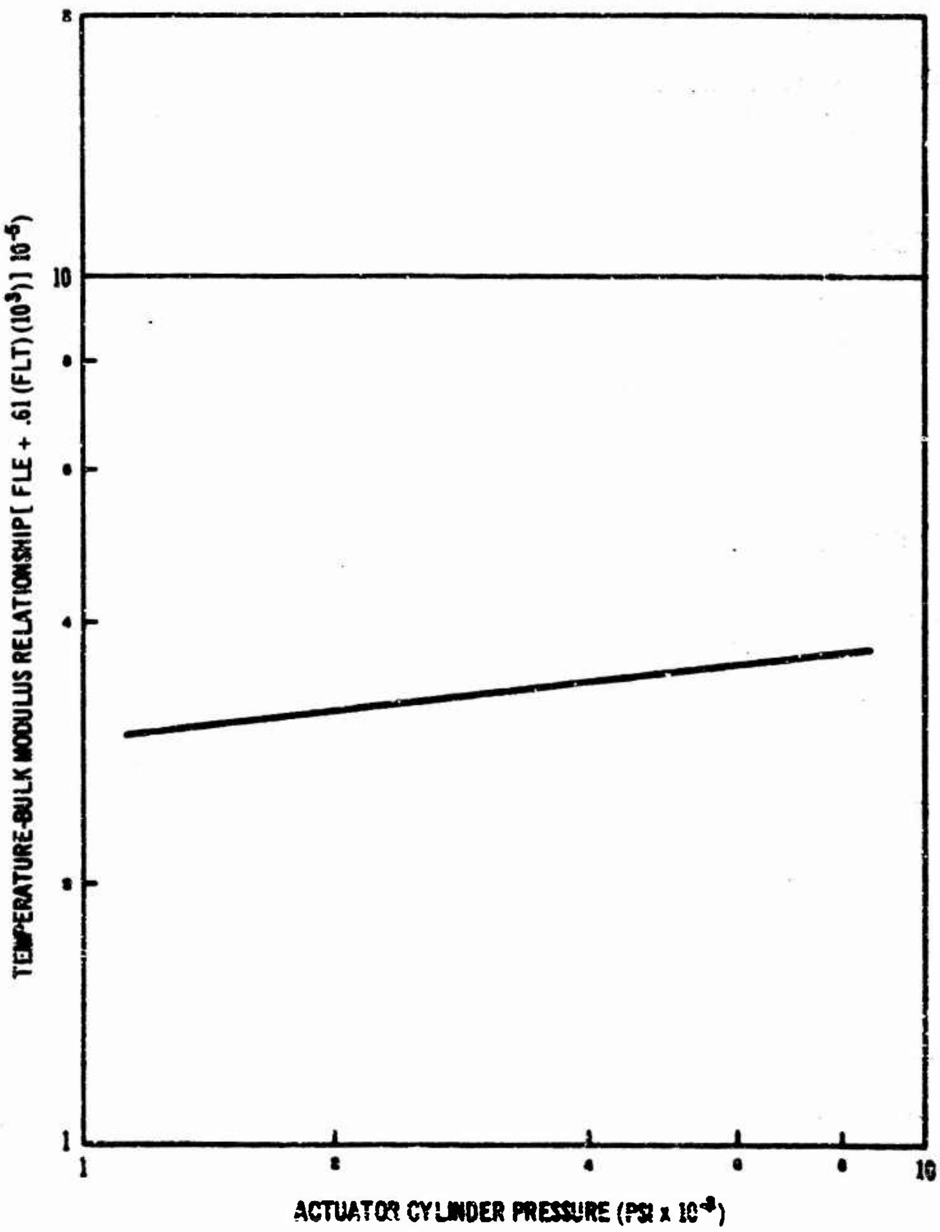


Figure 43. Adiabatic Bulk Modulus of MIL-H-5606A

In this expression, it was assumed that the actuator cylinder pressure at steady state and for small cyclic inputs can be approximated by a value of two-thirds of the system pressure.

11. ACTUATOR CYLINDER ELASTICITY

Table XIII summarizes the parameter relationships associated with the actuator cylinder elasticity. Derivations of these relationships follow the summary sheet.

Under static conditions, expansion of the actuator cylinder tends to lower cylinder pressure so it will be considered in determining the system elasticity. Cylinder expansion is a function of the following:

1. Cylinder pressure.
2. Piston area.
3. Cylinder thickness.
4. Piston rod diameter.
5. Material modulus of elasticity (E).
6. Poissons ratio.

Assuming that the last three factors are constant and using stress-strain relationships for a thick-walled steel cylinder, cylinder elasticity can be approximated by the straight-line relationship shown by Figure 44. Based on these data, the actuator elasticity for a steel cylinder as a function of system pressure and actuator area can be expressed by

$$\frac{1}{CYE} = 0.0019\,(AREA)^{-0.41}\,(PRES)^{-0.88}$$

12. MECHANICAL LINKAGE

Table XIV summarizes the parameter relationships associated with the mechanical linkage. Derivations of the relationships follow the summary sheet.

The mechanical linkage includes all of those parts used in the transmission of a mechanical signal from the cockpit to the valve actuator package for each elevator control surface. This includes cockpit control columns, cable tension regulators, torque override units, load feel mechanisms, and miscellaneous cabling and linkages. This mechanical control link will be considered as a complete assembly with weight being the only item of interest. Because none of the design parameters being considered influences the design of this control link, its weight will be considered constant at a value of

$$MLW = 108 \text{ lb}$$

TABLE XIII
ACTUATOR CYLINDER ELASTICITY SUMMARY SHEET

PAGE ______

NAME: ACTUATOR CYLINDER ELASTICITY

SYMBOL C Y E

INPUTS: A R E A
P R E S

OUPUTS C Y E

OUTPUT EQUATIONS

WEIGHT		
COST		
SIZE		
RELIABILITY		
ELASTICITY	C Y E	= AREA**.41*PRES**.88/.0019

NOTES:

TABLE XIV

MECHANICAL LINKAGE SUMMARY SHEET

PAGE ______

NAME: MECHANICAL LINKAGE

SYMBOL M L

INPUTS: N O N E

OUPUTS M L W

OUTPUT EQUATIONS

WEIGHT	M L W	= 108.
COST		
SIZE		
RELIABILITY		

NOTES: Data obtained from Aircraft Division, Douglas Aircraft Co., Long Beach, California.

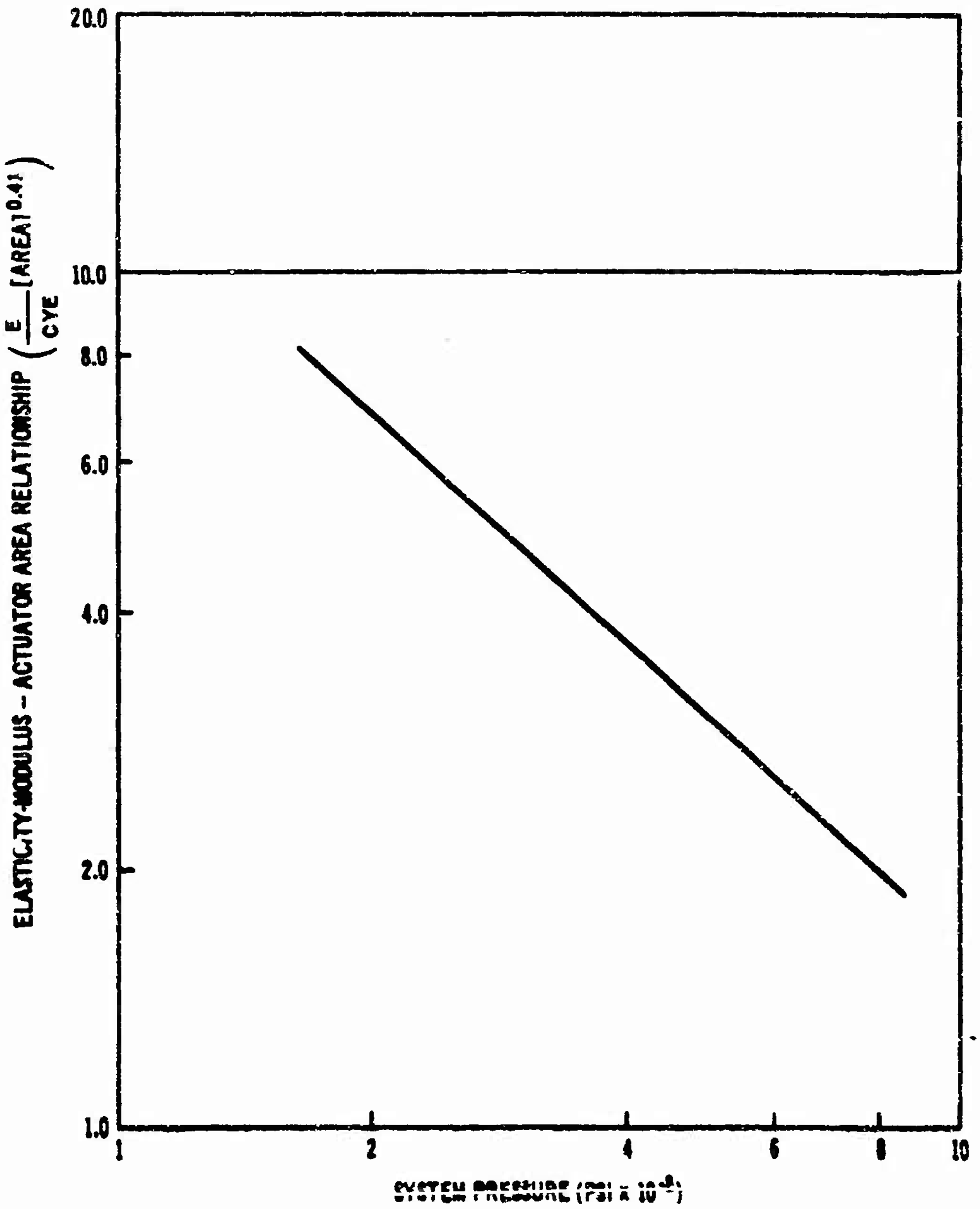


Figure 44. Actuator Cylinder Elasticity

APPENDIX II

DYNAMIC MODEL RELATIONSHIPS[1]

A hydraulic actuation system operates in a nonlinear way and so, logically, it should be represented by a nonlinear dynamic model. Such a model is shown in Figure 45. The model includes the essential characteristics of an actual hydraulic flow-control system. Those refinements, however, such as valve lap, valve leakage, and actuator torque-arm elasticity, which identify a particular system, have not been included for simplicity. The absence of these refinements influences model performance primarily in the resonant area; however, as shown in Figure 46, a portion of the characteristic resonant peak is still present. Data on a more complete model can be found in Reference 6.

When designing a control system, some form of electronic compensation normally is used to counteract the adverse effect of the resonant peak on system performance and to more or less linearize system response. In this design, a linearized response and a linearized model were desired to facilitate use of the ISE criterion but, as mentioned previously, the electronic portion of the control system was not included and so could not be used for this purpose. As a result, to retain the essential hydraulic system characteristics, and still have a fairly linear response, it was decided to add dynamic pressure feedback to the actuation system. For simplicity, the effect of the dynamic pressure feedback will be shown only on the final linearized model. Because the dynamic pressure feedback will not affect the equations derived in this section, its effect can be added to the final form by application of the superposition principle.

Referring to the nonlinear dynamic model (Figure 45), displacement of the valve spool, X, is the input to the system, and the actuator piston motion, Y_p, is the output. From the block diagram, flow into cylinder No. 1 is

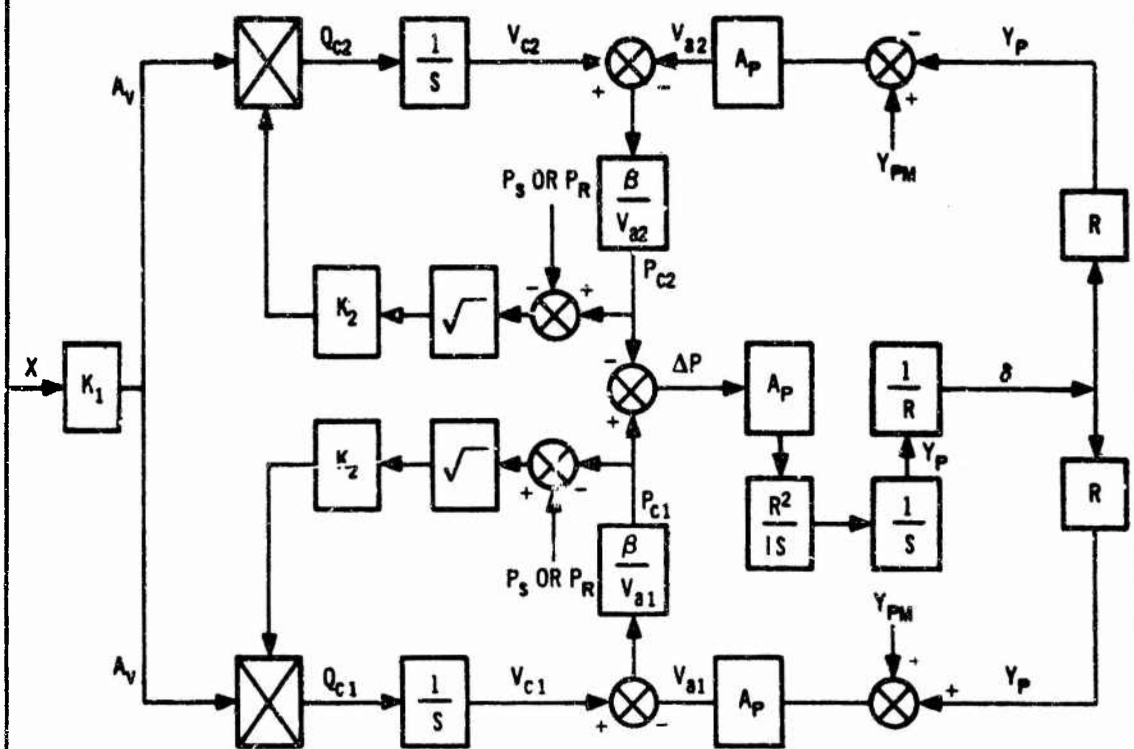
$$Q_{c1}(s) = K_1 K_2 \sqrt{P_s - P_{c1}(s)}\, X(s)$$

If X_n is defined as the nominal spool position, P_{cn} the nominal cylinder pressure, and small perturbations about these values are considered, flow is given as

$$\Delta Q_{c1}(s) = \frac{\partial Q_{c1}(s)}{\partial P_{c1}(s)} \Delta P_{c1}(s) + \frac{\partial Q_{c1}(s)}{\partial x(s)} \Delta X(s)$$

$$= -\frac{1}{2} K_1 K_2 \left(P_s - P_{c1}(s)\right)^{-1/2} x_n \Delta P_{c1}(s) + K_1 K_2 \left(P_s - P_{cn}\right)^{1/2} \Delta X(s)$$

[1] The symbols used in Appendix II have been simplified as an aid to the development of the required functions. All symbols used, however, have been defined in the appendix.



WHERE

- X – VALVE SPOOL MOTION
- K1 – VALVE AREA – STROKE RELATIONSHIP
- K2 – FLOW CONSTANT
- A_V – VALVE METERING AREA
- Q_C – VALVE FLOW
- V_C – UNCOMPRESSED FLUID VOLUME SUBJECT TO CYLINDER
- P_S, P_R – SUPPLY AND RETURN PRESSURE
- V_a – ACTUATOR VOLUME SUBJECT TO CYLINDER PRESSURE
- P_C – CYLINDER PRESSURE
- β – SYSTEM ELASTICITY
- A_P – ACTUATOR PISTON AREA
- I – LOAD INERTIA
- Y_P – ACTUATOR OUTPUT
- δ – LOAD OUTPUT
- Y_{PM} – MAXIMUM ACTUATOR OUTPUT
- ΔP – ACTUATOR PISTON DIFFERENTIAL PRESSURE
- R – ACTUATOR MOMENT OR TORQUE ARM
- S – OPERATOR d/dt

Figure 45. Nonlinear Dynamic Model

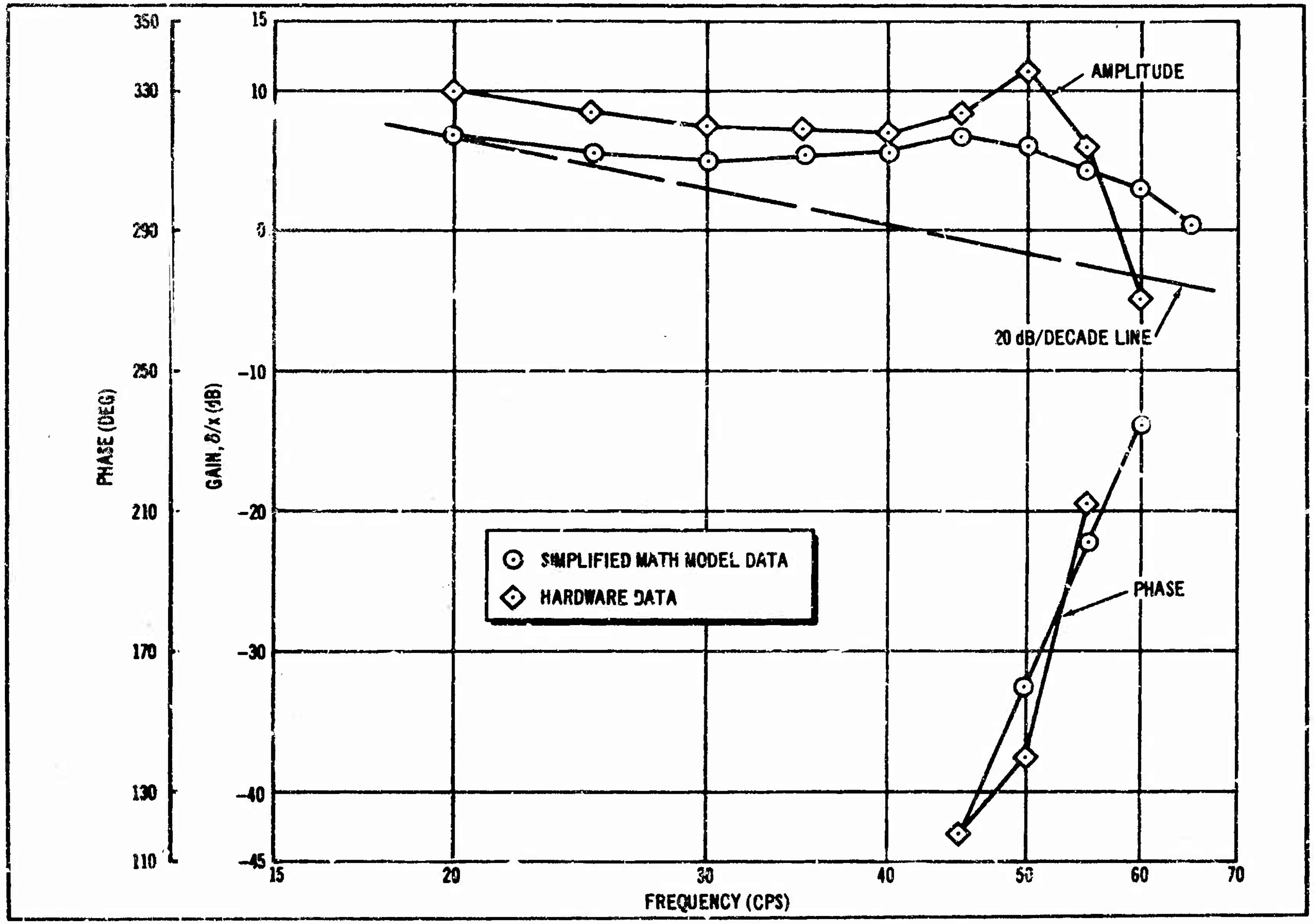


Figure 46. Valve-Actuator Open-Loop Frequency Response (10% Input)

The nominal conditions are

$$X_n = 0$$

$$P_{cn} = P_s/2$$

Then, the flow is

$$\Delta Q_{c1}(s) = K_1 K_2 \sqrt{P_s/2}\,\Delta X(s).$$

An identical relationship can be derived for Q_{c2}; it must be remembered, however, that the flows bear the opposite sign. From the block diagram, the cylinder pressure with a nominal cylinder volume Van, is given as

$$P_{c1}(s) = Q_{c1}(s)\left(\frac{\beta}{Van}\right)\frac{1}{s} - \frac{A_p\beta}{Van} Y_p(s)$$

and similarly for cylinder No. 2. The pressure drop across the actuator is

$$\Delta P(s) = \left[\frac{2K_1K_2\sqrt{P_s/2}\,\beta}{Van}\right]x(s) - \frac{2\beta A_p}{Van} Y_p(s)$$

Making the following definitions

$$K_q = \frac{2K_1K_2\sqrt{P_s/2}\,\beta}{Van}$$

and

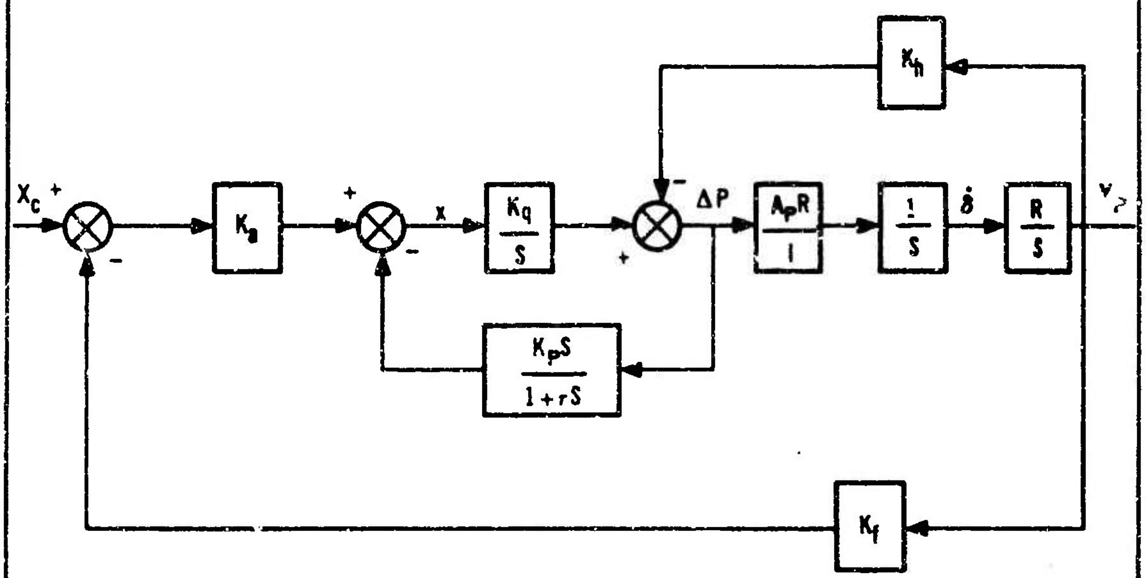
$$K_h = \frac{2\beta A_p}{Van}$$

Therefore, the pressure drop can be written as

$$\Delta P = \frac{K_q}{s} X(s) - K_h Y_p(s)$$

Combining this with the representation of the dynamic pressure feedback given by Moog (Reference 7) the block diagram shown in Figure 47 describes the linearized system.

To verify accuracy of the linearized model, a frequency response of the linear model was compared with the frequency response of the nonlinear model taken at different input levels. The results for a 10% input is shown in Figure 48. As is readily seen from these data, the frequency response curves of the nonlinear model with dynamic-pressure feedback and the linear model compare well. From additional data taken, but not shown, good comparison was found for input levels up to 50%.



WHERE

X_c	– COMMAND INPUT
X	– VALVE SPOOL MOTION
K_a	– AMPLIFIER AND TORQUE MOTOR GAIN
K_q	– FLOW CONSTANT
K_p	– PRESSURE FEEDBACK GAIN
τ	– TIME CONSTANT OF PRESSURE FEEDBACK
K_f	– FEEDBACK GAIN
K_h	– EQUIVALENT HYDRAULIC SPRING
ΔP	– ACTUATOR PISTON DIFFERENTIAL PRESSURE
A_p	– ACTUATOR PISTON AREA
I	– LOAD INERTIA
Y_p	– ACTUATOR OUTPUT
$\dot{\delta}$	– LOAD VELOCITY
R	– ACTUATOR MOMENT OR TORQUE ARM
S	– OPERATOR d/dt

Figure 47. Linearized Dynamic Model

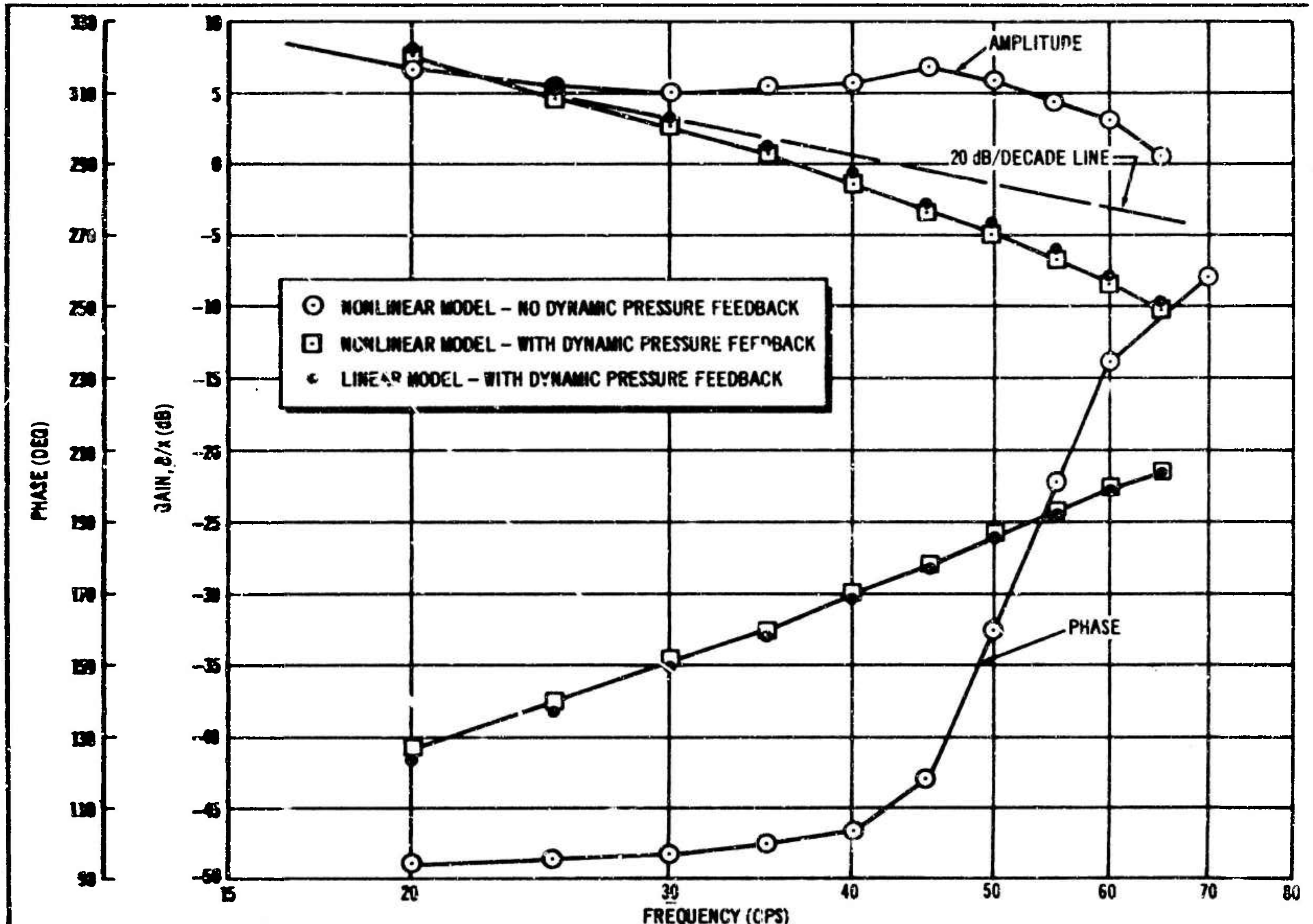


Figure 48. Valve-Actuator-Load Open-Loop Frequency Response

APPENDIX III

NO. 1 COMPUTER PROGRAM

1. DESCRIPTION OF THE FIXED-GRID COMPUTER PROGRAM

The computer program was written with a series of subroutines for the evaluation of the cost functions. This was done for ease of understanding the completed program and for convenience in making corrections and additions. The entire program is written in Fortran IV language. The main program basically sets up the grid over which the performance index will be evaluated and calculates the performance index. There are six subroutines: one each for the five cost functions and one for the evaluation of the nominal system. The fixed grid is used to evaluate in the following manner:

a. The program fixes a pressure, moment arm, and the tube sizes, and then evaluates the minimum area that will meet the hinge-moment requirements. The performance index and cost functions will be evaluated for these parameter values.

b. Then holding all of the parameters fixed, the area will be increased by 0.5 sq in. and the performance criterion re-evaluated. This will be repeated three times.

c. Then the pressure will be increased by adding a fixed, predetermined amount (250 psi for this example), then the minimum area will again be calculated, and the performance index evaluated. The area will then be incremented as indicated in step b.

d. After the entire range of pressure has been investigated at the initial moment arm and tube diameters, the moment arm will be decreased by a fixed constant (0.5 in. for this example). Then the entire range of pressures and the corresponding areas will be re-evaluated. This will continue until all of the moment-arm values to be investigated have been used.

e. After this has been accomplished for the initial values of the tubes, the entire process must be repeated again for each possible combination of the tube sizes. This will cover the parameter space.

The logic for each subroutine is similar. The flow diagram for the weight subroutine shown by Figure 49 is typical of this logic.

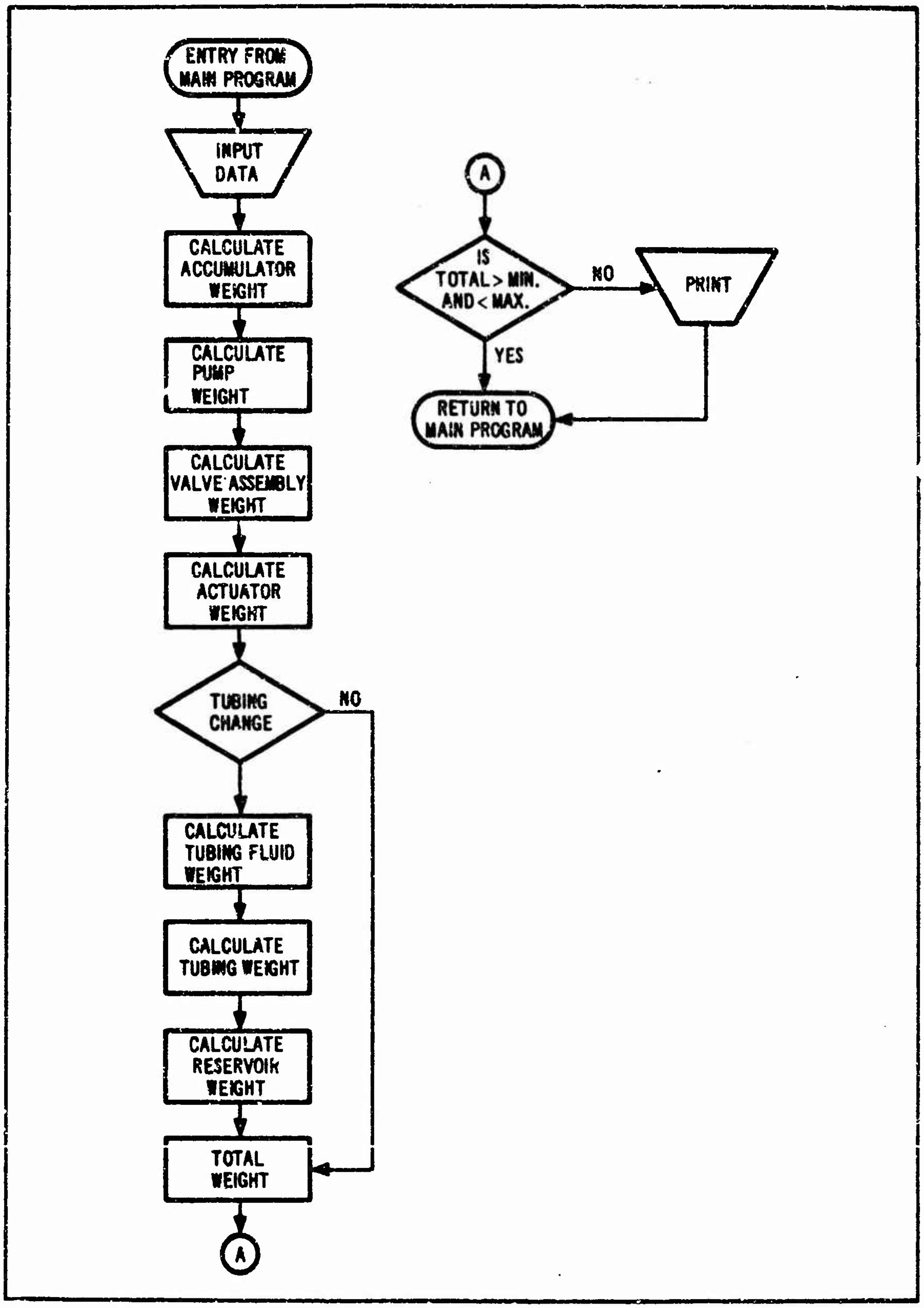


Figure 49. Weight Subroutine Logic

2. COMPUTER VARIABLES NOT PREVIOUSLY DEFINED

Computer variables not previously defined and used in the No. 1 listing include the following:

TOTW	total weight.
TNOMW	total nominal weight.
TOTC	total cost.
TNOMC	total nominal cost.
TOTDYP	total dynamic performance.
TOTV	total volume.
TNOMV	total nominal volume.
TOTR	total reliability.
TNOMR	total nominal reliability.
VEQ	equivalent volume.
VEQN	nominal equivalent volume.
REDN	nominal reservoir diameter.
RELN	nominal reservoir length.
ACTLN	nominal actuator length.
WR	desired bandwidth of actuation system.

LISTING FOR NO. 1 COMPUTER PROGRAM

```
-IT FOR MAIN
C     OPTIMAL DESIGN OF AN ACTUATING SYSTEM                                     MAIN0000
      DIMENSION  PARAM(31), FJS(6)                                              MAIN0003
      COMMON   /DATA/ A(38)                                                     MAIN0006
      COMMON   /NOMI/  TNOMW ,TNOMC,TNOMV,THOMR,TNOMDY ,TNOMEN                  MAIN0009
      COMMON   /OUP/   TOTW,TOTC,TOTV,TOTR,TOTDYP,TOTENV,ACTW                   MAIN0012
      COMMON   /FLG/ IFPFLG,IERFLG,IDIA,NNEND,NN                                MAIN0015
      COMMON /INTM/ DS1,DS2,DAL1,DAL2, Q, PRES,AREA,R,ACTDP,TUFLW,ANORM         MAIN0018
      EQUIVALENCE ( A(26),TULAL1),( A(27),TULAL2),(A(28),TULS1),                MAIN0019
     1 (A(29),TULS2)                                                            MAIN0020
      WRITE (6,553)                                                             MAIN0023
      READ (5,500) (A(I),I=1,38)                                                MAIN0024
      WRITE (6,550)  (A(I),I=1,38)                                              MAIN0027
      READ (5,501)  NFND, MEND                                                  MAIN0030
      WRITF (6,551)  NEND, MEND                                                 MAIN0033
      WRITE (6,555)                                                             MAIN0034
      DO 10  II= 2,31                                                           MAIN0036
   10 PARAM(II) = 0.0                                                           MAIN0039
      IFPFLG = 0                                                                MAIN0042
      IERFLG = 0                                                                MAIN0045
      NNEND = 5                                                                 MAIN0069
      IEND = 1                                                                  MAIN0072
      JEND = 1                                                                  MAIN0075
      KEND = 1                                                                  MAIN0078
      LEND = 1                                                                  MAIN0081
      IF( A(7).NE. 0.0)  LEND = 2                                               MAIN0084
      IF( A(9).NE. 0.0)  KEND = 2                                               MAIN0087
      IF( A(11).NE. 0.0) JEND = 2                                               MAIN0090
      IF( A(13).NE. 0.0) IEND = 2                                               MAIN0093
      PARAM(1) = A(3)                                                           MAIN0096
      DO15 II=2,9
   15 PARAM(II) =  PARAM(II-1) + A(5)                                           MAIN0102
      PARAM(1)=5.5
      PARAM(10) = A(2)
      DO18 II=2,17
   18 PARAM(II+9) = PARAM(II+8) + A(4)
      PARAM(10)= A(1)
      DO 200  I= 1,IEND                                                         MAIN0114
      DO 200  J= 1,JEND                                                         MAIN0117
      DO 200  K= 1,KEND                                                         MAIN0120
      DO 200  L= 1,LEND                                                         MAIN0123
      DO 200  M= 1,MEND                                                         MAIN0126
      DO 200  N= 1,NEND                                                         MAIN0129
      DO 199 NN= 1,NNEND                                                        MAIN0132
      PRES = PARAM( N+9)                                                        MAIN0159
      R = PARAM(M)                                                              MAIN0162
      AREA  = AREA + .5                                                         MAIN0165
      DAL1  = A(L+5)                                                            MAIN0168
      DAL2  = A(J+9)                                                            MAIN0171
      DS1   = A(K+7)                                                            MAIN0174
      DS2   = A(I+11)                                                           MAIN0177
      CALL NOMSYS                                                               MAIN0135
      IF( IDIA .EQ. 0)   GO TO 25                                               MAIN0136
      ZZ3 = PRES - 3000.
      IF(ZZ3.LT.0.0) ZZ3 =0.0
      ZZ1 = .3366*( TULS1*(DS1)**(-3.98)*(1.+.0002*ZZ3)**2 + TULAL1*(DAL
     11)**(-3.98))
      ZZ2 = .3366*( TULS2*(DS2)**(-3.98)*(1.+.0002*ZZ3)**2 + TULAL2*(DAL
     12)**(-3.98))
   25 TUDP4    =  Q* ( 2.2*ZZ1 + ZZ2)                                           MAIN0136
      IF ( PRES .GE. TUDP4  )  GO TO 30                                         MAIN0136
```

```
      WRITE (6,554) I,J,K,L,M,N,NN,TUDP4                               MAIN0156
      IF (IFPFLG .EQ. 0) GO TO 210                                     MAIN1136
      GO TO 199                                                        MAIN0137
   30 CALL WEIGHT                                                      MAIN0138
      IF( IERFLG .EQ. 0) GO TO 35                                      MAIN0141
   32 PIDX =  0.0                                                      MAIN0144
      IERFLG = 0                                                       MAIN0147
      IF (IFPFLG .EQ. 0) GO TO 210                                     MAIN0148
      GO TO  115                                                       MAIN0150
   35 CALL COST                                                        MAIN0150
      IF ( IERFLG .NE. 0) GO TO 32                                     MAIN0152
      CALL DYNPER                                                      MAIN1152
      IF ( IERFLG .NE. 0) GO TO 32                                     MAIN2152
      CALL VOLUME
      IF ( IERFLG .NE. 0) GO TO 32
      CALL  RELIAB
      IF ( IERFLG .NE. 0 ) GO TO 32
      IF (IFPFLG.NE. 0) GO TO 110                                      MAIN0139
      PARAM(1)=A(3)
      PARAM(10)= A(2)
      IFPFLG = 1                                                       MAIN0140
  110 PIDX = 0.0                                                       MAIN0153
      FJS(1) = 1.0 - TOTW/(2.0*TNOMW)                                  MAIN0154
      FJS(2) = 1.0 - TOTC/(2.0*TNOMC)                                  MAIN1154
      FJS(3) = 1. - TOTDYP
      FJS(4)= 1.0-TOTV/(2.0*TNOMV)
      FJS(5) = 1.0 - TOTR/ (2.0 * TNOMR)
      DO  112  II = 1,5                                                MAIN6154
      IF ((FJS(II).LT. 0.0).OR.(FJS(II).GT.1.0)) GO TO 115             MAIN7154
  112 CONTINUE                                                         MAIN8154
      DO  114  II = 1,5                                                MAIN9154
  114 PIDX =  PIDX +  A(II+31)*FJS(II)                                 MAIN0155
  115 WRITE  (6,552)  I,J,K,L,M,N,NN,PIDX,TOTW,TOTC,TOTDYP,TOTV,TOTR,  MAIN0156
     1ACTW
  199 CONTINUE                                                         MAIN0180
      NNEND = 4                                                        MAIN0189
  200 CONTINUE                                                         MAIN0192
  210 STOP                                                             MAIN0195
  500 FORMAT ( 7F10.3 )                                                MAIN0198
  501 FORMAT ( 2(3X,I2) )                                              MAIN0201
  550 FORMAT ( 1H0, 47X,'JHINPUT DATA////13H P NOM     = F10.3,11H P MINMAIN0204
     1    = F10.?,10H R NOM  = F10.3,10H P INC  = F10.3,11H R INC    = MAIN0207
     2F10.3//13H DAL11      = F10.3,11H DAL12   = F10.3,10H DS11    =  MAIN0210
     3F10.3,10H DS12   = F10.3,11H DAL21    = F10.3,11H DAL22    = F10.3//MAIN0215
     413H DS21       = F10.3,11H DS22     = F10.3,10H W MAX  = F10.3,10H WMAIN0216
     5 MIN  = F10.3,11H C MAX    = F10.3,11H C MIN    = F10.3//13H PUV MAXMAIN0219
     6   = F10.3,11H RED MAX = F10.3,10H R MAX  = F10.3,10H R MIN  =   MAIN0222
     7F10.3,11H DYP MAX = F10.3,11H DYP MIN = F10.3//13H ENV MAX    =  MAIN0225
     8F10.3,11H ENV MIN = F10.3,10H TULAL1 = F10.3,10H TULAL2 = F10.3, MAIN0228
     911H TULS1    = F10.3,11H TULS2    = F10.3//13H DELTA MAX = F10.3, MAIN0231
     111H HM       = F10.3,10H WTF1   = F10.3,10H COSTF2 = F10.3,11H DYPFMAIN0232
     23    = F10.3//13H VOLF4      = F10.3,11H RELF5    = F10.3,10H ENVF6 MAIN1232
     3= F10.3,11H REL MAX = F10.3)                                     MAIN3232
  551 FORMAT ( 1H0, 11H P STATES = I4,5X,10HR STATES = I4)             MAIN0234
  552 FORMAT (1H , 6H CASE 5I1,I2,I1,3X,7HP.I. = F7.2,3X,
     15HWT = F8.1,3X,7HCOST = F7.0,3X,6HDYP = F6.2,3X,6HVOL = F6.2,3X, MAIN0240
     26HREL = F7.1,3X,7HACTW = F7.2)                                   MAIN1240
  553 FORMAT (1H1, 40X, 24HOPTIMAL ACTUATING SYSTEM  )                 MAIN0241
  554 FORMAT ( 1H , 6H CASE      I1,I1,I1,I1,I1,I2,I1,5X, 8HTUDP4 =    MAIN0242
     1 F10.3)                                                          MAIN0243
  555 FORMAT(1H1,34X,36HCASE = DS2,DAL2,DS1,DAL1,R,PRES,AREA )          MAIN1243
```

```
      END                                                               MAIN0244
```

```
-IT FOR NOMSYS
      SUBROUTINE  NOMSYS
      COMMON /DATA/ A(25),TULAL1,TULAL2,TULS1,TULS2,DELMX,HM
      COMMON /FLG/ IFPFLG,IERFLG,IDIA ,NNEND,NN
      COMMON /INTM/  DS1,DS2,DAL1,DAL2,C,PRES,AREA,R,ACTDP,TUFLW,ANORM
      IF ( IFPFLG.NE.0) GO TO 3
      ZZ5 =   .004533*DELMX
      DS10 = DS1
      DS20 = DS2
      DAL10 = 0.0
      DAL20 = 0.0
      GO TO 5
    3 IF (NN .EQ. 1)  GO TO 5
      IF((NNEND.EQ.5).AND.(NN.EQ.2)) GO TO 5
      GO TO 15
    5 AREA = HM/(PRES* R)
      ANORM = AREA
      IF (DAL1.EQ.DAL10) GO TO 7
      DAL10 = DAL1
      GO TO 10
    7 IF ( DS1.EQ.DS10) GO TO 8
      DS10 = DS1
      GO TO 10
    8 IF ( DAL2.EQ.DAL20) GO TO 9
      DAL20 = DAL2
      GO TO 12
    9 IF ( DS2.EQ.DS20) GO TO 15
      DS20 = DS2
      GO TO 12
   10 IDIA = 1
      ZZ6 = PRES - 3000.
      IF(ZZ6.LT.0.0) ZZ6 = 0.0
      ZZ1 =   .001705*( TULS1*(DS1)**(-4.93)*(1.+.0003*ZZ6) + TULAL1*(DAL
     11)**(-4.93))
      GO TO 8
   12 IDIA = 1
      ZZ2 =   .001705*( TULS2*(DS2)**(-4.93)*(1.+.0003*ZZ6) + TULAL2*(DAL
     12)**(-4.93))
   15 Q = ZZ5*AREA*R
      ZZ3 =   (4.4* Q)**1.75
      TUDP   = ZZ1*(8.8*Q)**1.75 + ZZ2*ZZ3
      TUDP2  = ZZ1* ZZ3 + ZZ2 *(2.2*Q)**1.75
      ACTDP = PRES - .25*( PRES - TUDP ) - TUDP2
      IF (IFPFLG .NE. 0)  GO TO 25
      ZZ4 = ( .75*PRES - ACTDP)
      IF (ZZ4.LE.100.0) GO TO 20
      PRES = PRES + ZZ4/ 2.0
      GO TO 5
   20 WRITE (6, 30)  PRES,AREA, Q, TUDP,TUDP2,ACTDP
   25 RETURN
   30 FORMAT(1H0, 8H PRES = F10.3  8H AREA = F10.3, 5H Q = F10.3,
     1 9H TUDP  = F10.3,10H TUDP2  = F10.3, 9H ACTDP = F10.3)
      END
```

```
-IT FOR WEIGHT
      SUBROUTINE  WEIGHT
      COMMON /DATA/ A(13),WMX,WMN, DUM(10),TULAL1,TULAL2,TULS1,TULS2,
     1 DELMX, HM
      COMMON /FLG/ IFPFLG,IERFLG,IDIA
      COMMON /INTM/ DS1,DS2,DAL1,DAL2,Q,PRES,AREA,R,ACTDP,TUFLW,ANORM
      COMMON /OUP/ TOTW,DUMA(5),ACTW
      COMMON /NOMI/ TNOMW
      IF (IFPFLG.NE.0) GO TO 5
      ZZ2 =ABS(SQRT(0.0625*(1.+4.*(5.0-56.*3000.**(-0.280))**2)))+140.*
     13000.**(-0.242)
      ACCW = 3. + (PRES - 1000.)*0.001
    5 ZZ3 = PRES - 3000.0
      ZZ4 = ZZ3
      IF( ZZ4.LT. 0.0) ZZ4 = 0.0
      PUW =  (7.5 + 5.412*Q)* (1.0+.00005* ZZ4)
      VASW = (31.5 + 3.5*Q)*( 1.0 + .00005*ZZ3)
      IF ( IDIA.EQ.0) GO TO 10
      ZZ5  = (1.6* DS1 - .489)*TULS1 + (1.6* DS2 - .489)*TULS2
      ZZ6  = ( .695*DAL1 - .1855)*TULAL1 + ( .695*DAL2 - .1855)*TULAL2
      TFWS1 = .293*TULS1*(.430 + .88*(DS1-.5))**2
      TFWS2=.293*TULS2*(.430+.88*(DS2-.5))**2
      TFWAL2=.293*TULAL2*(.430+.88*(DAL2-.5))**2
      TFWAL1=.293*TULAL1*(.430+.88*(DAL1-.5))**2
      TUFLW = (TFWS1 +TFWS2)*(1. + 2.4*10.**(-9.)*(ZZ4)**2) + TFWAL1 + T
     1FWAL2
      REW = 1.8 + .219*TUFLW
   10 ACTW=.96*(ZZ2+3.3*(AREA-3.20)+.95*(PRES-3000.)/1000.+1.20*(R-5.0))
      TUW =  ZZ5* (1.0 + .00012* ZZ4) + ZZ6
      XMLW = 108.
      TOTW=PUW +ACCW+ 4.0*VASW + TUW + REW + 4.0*ACTW + XMLW
      IF (IFPFLG.EQ. 0) TNOMW  = TOTW
      IF ( (TOTW.LE.WMX).AND.(TOTW.GE.WMN))  GO TO 15
      IERFLG = 1
      WRITE (6,25)
   15 RETURN
   25 FORMAT (1H , 10X, 20HWEIGHT OUT OF LIMITS )
      END
```

```
-IT FOR COST
      SUBROUTINE COST
      COMMON /DATA/ A(15),CMX,CMN,DUM(8),TULAL1,TULAL2,TULS1,TULS2
      COMMON /FLG/ IFPFLG,IERFLG,IDIA
      COMMON /INTM/ DS1,DS2,DAL1,DAL2,Q,PRES,AREA,R,ACTDP,TUFLW
      COMMON /OUP/ TOTW,TOTC
      COMMON /NOMI/ TNOMW,TNOMC
      IF (IFPFLG.NE. 0) GO TO 5
      ACCC = 150.0
      VAPC = 7500.0
    5 PUC = 1400.0 + 123.2* Q
      IF ( IDIA .EQ. 0) GO TO 10
      IDIA = 0
      REC = 245.0 +  .91* TUFLW
      ZZ1 = 1.3 *( DS1*TULS1 + DS2*TULS2)
      ZZ2 =   .222 *( DAL1*TULAL1 + DAL2*TULAL2)
   10 ZZ3 = PRES - 3000.0
      IF ( ZZ3 .LT. 0.0)  ZZ3 = 0.0
      TUC = ZZ1*( 1.0 + .00013*ZZ3) + ZZ2
      TOTC = ACCC + VAPC + PUC + REC + TUC
      IF ( IFPFLG .EQ. 0)  TNOMC = TOTC
      IF ( (TOTC.LE.CMX).AND.(TOTC.GE.CMN)) GO TO 15
      IERFLG = 1
      WRITE (6,25)
   15 RETURN
   25 FORMAT (1H ,10X, 18HCOST OUT OF LIMITS )
      END
```

```
-IT FOR DYNPER
      SUBROUTINE DYNPER                                                 DYNP0000
      COMMON /DATA/ DUM(21),DYPMX,DYPMN                                 DYNP0003
      COMMON /FLG/ IFPFLG, IERFLG                                       DYNP0006
      COMMON /INTM/ DUM2(5),PRES,AREA,R
      COMMON /OUP/  DUM3(4),TOTDYP                                      DYNP0012
      COMMON /NOMI/ DUM4(4),TNOMDY                                      DYNP0015
      WR = 10.
      CYE = AREA **0.41*PRES**0.88/0.0019
      FLT = 120.
      FLE = -610.*FLT+138500.*(2.*PRES/3.)**.1082
      SE  = FLE * CYE/(FLE + CYE)
      SIGM = 40./57.3
      AYE = 420.
      WN = SQRT (2.* SE *AREA*R/(SIGM * AYE))
      TOTDYP = 1.-EXP(-.25*(WR-0.1*WN))
      IF(TOTDYP.LT.0.0) TOTDYP = 0.0
      IF(TOTDYP.GT.1.0) TOTDYP = 1.0
   12 IF (IFPFLG.EQ. 0)  TNOMDY = TOTDYP                                DYNP0024
      IF ( (TOTDYP .LE. DYPMX).AND.(TOTDYP .GE. DYPMN))  GO TO 15       DYNP0027
      IERFLG = 1                                                        DYNP0030
      WRITE (6,20)                                                      DYNP0033
   15 RETURN                                                            DYNP0036
   20 FORMAT (1H , 10X, 23HDYN. PER. OUT OF LIMITS)                     DYNP0039
      END                                                               DYNP0042
```

```
-I1 FOR VOLUME
      SUBROUTINE VOLUME                                                  VOLU0000
      COMMON /DATA/ A(17), PUVMX, REDMX, DUM(10), DELMX, DUM1(7),RELMX   VOLU0003
      COMMON /FLG/  IFPFLG, IERFLG                                       VOLU0006
      COMMON /INTM/ DUM2(6), Q,PRES,AREA,R,ACTDP,TUFLW                   VOLU0009
      COMMON /OUP/  TOTW,TOTC,TOTV                                       VOLU0012
      COMMON /NOM1/ TNOMW,TNOMC,TNOMV                                    VOLU0015
      IF ( IFPFLG .NE. 0) GO TO 5                                        VOLU0018
      ZZ1 = .057356 * DELMX                                              VOLU0021
    5 PUV = 40.0 + 199.7 * Q                                             VOLU0024
      IF ( PUV .LE. PUVMX)  GO TO 10                                     VOLU0027
      WRITE ( 6,50) PUV                                                  VOLU0030
      GO TO 18                                                           VOLU0033
   10 VEQ =  PUV + 90.00                                                 VOLU0036
      RED =  6.0 + .04 * TUFLW                                           VOLU0039
      IF ( RED .LE. REDMX)  GO TO 15                                     VOLU0042
      WRITE  (6,51) RED                                                  VOLU0045
      GO TO 18                                                           VOLU0048
   15 REL = 9.0 +  .18 * TUFLW                                           VOLU0051
      IF ( REL .LE. RELMX) GO TO 20                                      VOLU0054
      WRITE  (6,52) REL                                                  VOLU0057
   18 IERFLG  = 1                                                        VOLU0060
      TOTV = 0.0                                                         VOLU0063
      RETURN                                                             VOLU0066
   20 ACTL =  ZZ1 * R + 13.0                                             VOLU0069
      IF ( IFPFLG .NE. 0)  GO TO 21                                      VOLU0072
      VEQN = 2.0 * VEQ                                                   VOLU0075
      REDN = 2.0 * RED                                                   VOLU0078
      RELN = 2.0 * REL                                                   VOLU0081
      ACTLN = 2.0 * ACTL                                                 VOLU0084
   21 IF(5.6 - R) 22,23,24
   22 SGN = -1.
      GO TO 25
   23 SGN = 0.
      GO TO 25
   24 SGN = 1.
   25 TOTV = VEQ/VEQN + RED/REDN + REL/RELN + ACTL/ACTLN + .125*(1.-SGN)
      IF (IFPFLG .EQ. 0)  TNOMV = TOTV                                   VOLU0088
      RETURN                                                             VOLU0090
   50 FORMAT (1H , 10X,  6HPUV = F8.2, 12H EXCEEDS MAX )                 VOLU0093
   51 FORMAT (1H , 10X,  6HRED = F8.2, 12H EXCEEDS MAX )                 VOLU0096
   52 FORMAT (1H , 10X,  6HREL = F8.2, 12H EXCEEDS MAX )                 VOLU0099
      END
```

```
-IT FOR RELIAB
      SUBROUTINE  RELIAB                                                 REL10000
      COMMON /DATA/ DUM(19), RMAX                                        REL10003
      COMMON /FLG/  IFPFLG, IERFLG                                       REL10006
      COMMON /INTM/ DUM1(4), Q, PRES                                     REL10009
      COMMON /OUP/ DUM2(3), TOTR                                         REL10012
      COMMON /NOMI/ DUM3(3),TNOMR                                        REL10015
      IF (IFPFLG .NE. 0)  GO TO 20                                       REL10018
      ACCR = 12.3                                                        REL10021
      RER = 16.0                                                         REL10024
      TUR = 44.0                                                         REL10027
      VAPR = 16.0                                                        REL10030
      PTOTR = ACCR + RER + TUR + VAPR                                    REL10033
   20 PUR =  465.0 + 0.0111 * PRES * Q                                   REL10036
      TOTR = PTOTR + PUR                                                 REL10039
      IF  ( IFPFLG .EQ. 0)  TNOMR = TOTR                                 REL10042
      IF ( TOTR .LE. RMAX)  GO TO 30                                     REL10045
      IERFLG = 1                                                         REL10048
      WRITE (6,40)                                                       REL10051
   30 RETURN                                                             REL10054
   40 FORMAT (1H ,10X,23HRELIABILITY EXCEEDS MAX )                       REL10057
      END                                                                REL10060
```

```
- XQT MAIN
 3000.0     2000.0        7.5       250.0       -0.5         1.0        0.75   DATA0001
    1.0       0.75       0.75       0.625       0.75       0.625  99999.999   DATA0002
    0.0  99999.999        0.0   99999.999  99999.999   99999.999        0.0   DATA0003
   40.0        0.0        0.0         0.0      185.0        40.0      185.0   DATA0004
   40.0       40.0    48000.0        30.0       15.0        25.0       25.0   DATA0005
    5.0        0.0  99999.999                                                 DATA0006
   17    8
```

APPENDIX IV

NO. 2 COMPUTER PROGRAM

1. RANDOM AND ADAPTIVE RANDOM SEARCH

Before proceeding to the details of the computer mechanization, a brief mathematical description of the searching techniques is given. More details are presented in Reference 8.

2. RANDOM OPTIMIZATION METHOD

The random optimization method will be described, using the notation in the statement of the problem in Section III. If $\underline{u}(1)$ denotes the initial parameter set of the system, the performance index corresponding to this state will be denoted by P. I. (1). Let $\underline{\xi}$ be an n-dimensional normal random vector with zero mean value and unit correlation matrix. Compute the P. I. at the random parameter vector $\underline{u}(1) + \underline{\xi}(1)$ and call the corresponding random step a success if

$$\text{P. I. } [\underline{u}(1) + \underline{\xi}(1)] > \text{P. I. } (1) + \underline{\epsilon}$$

for an ϵ chosen *a priori*. Otherwise, the random step is called a failure. Depending on whether the first random step is a success or failure, the parameter vector $\underline{u}(2)$ is defined as follows:

$$\underline{u}(2) = \underline{u}(1) + \underline{\xi}(1) \quad \text{for a success}$$

$$\underline{u}(2) = \underline{u}(1) \quad \text{for a failure}$$

Next, the performance criterion is computed at the random parameter vector $\underline{u}(2) + \underline{\xi}(2)$, and the procedure is repeated. The recursive relationship at the k^{th} random step is seen to be: (1) compute P. I. $[\underline{u}(k) + \underline{\xi}(k)]$, (2) is P. I. $[\underline{u}(k) + \underline{\xi}(k)]$ > P. I. $[\underline{u}(k)] + \underline{\epsilon}$, (3) if step 2 is a success, define $\underline{u}(k + 1) = \underline{u}(k) + \underline{\xi}(k)$, if not $\underline{u}(k + 1) = \underline{u}(k)$. By continuing this process, a random sequence of states $\underline{u}(k)$ is obtained. The convergence of this sequence to the optimal state is guaranteed.

3. ADAPTIVE RANDOM OPTIMIZATION METHOD

The adaptive random optimization method is essentially the same as the method just described. To speed the convergence to the optimum system, however, the random vector is chosen in a slightly different manner. The mean of the random vector and its variance will be varied according to past experience of the optimization procedure; that is, define a random process

$$\underline{\zeta}(k) = \underline{d}(k) + T(k)\,\underline{\xi}(k)$$

where $\underline{d}(k)$ is the mean value of $\underline{\zeta}(k)$, and T(k) is a transformation matrix operating on the random vector $\underline{\xi}(k)$.

Once the random process $\underline{\zeta}(k)$ is defined, it is used in the same manner as the random vector was used in the random search method. All that remains is to describe calculation of the random process. If the first step is to be taken in a truly random manner, the initial value of the mean value vector should be zero. If the initial step was a success, then the mean value of the second step is chosen to bias the choice of the random step in the same direction.

That is,

$$\underline{d}(2) = c_1\underline{\zeta}(1).$$

If the initial step was a failure, the mean value of the second step remains zero or is chosen so as to ensure motion in the opposite direction. The recursion relation for the k^{th} random step is defined to be

$$\underline{d}(k + 1) = c_0\underline{d}(k) + c_1\underline{\zeta}(k)$$

where c_0 and c_1 satisfy the following conditions

$$0 \leq c_0 < 1, \ c_1 \geq 0, \ c_0 + c_1 > 1$$

when

$$PI(k) > PI(k - 1) + \underline{\epsilon}$$

$$0 \leq c_0 < 1, \ c_1 < 0, \ /c_0 + c_1/ < 1$$

when

$$PI(k) < PI(k - 1) + \underline{\epsilon}.$$

Updating the mean value in the manner described above increases the probability of a successful step, because it coincides with the direction of expected increase of the performance index. It also lengthens the random step and speeds up the process of finding the optimal system. Douglas has experienced success in applying this technique and has developed an algorithm for updating variance of the random process similar to the technique described above for updating the mean value. The results of this work are discussed in References 1 and 2.

4. DESCRIPTION OF THE ADAPTIVE RANDOM SEARCH ROUTINE

The computer program used essentially the same format that was used in the fixed-grid program. The subroutines used to evaluate the cost functions were identical. The main program was changed to incorporate the random search technique. In addition, it was found to be simpler to incorporate the design of the nominal system into the main program.

The flow diagram shown in Figure 13 shows the logic of the routine. The listing that follows (No. 2 Program) is only for the main program and shows how the mathematical description of the adaptive random search routine was incorporated into the program. Random numbers were selected for all of the independent variables (pressure, area, moment arm, and four tube sizes). Given this selection, this set had to be tested to ensure that it was compatible with the design requirements, such as, static hinge moment, flow at -40°F, and dynamic hinge moment. If all of the tests were passed, the parameter set was used to calculate the P. I. ; if not, a new set of parameters were selected and the process repeated. The old set was classed as unsuccessful for the purpose of updating the adaptive feature of the routine.

Unsuccessful trials were counted at three places in the program: (1) if the hinge-moment test (ARTEST) failed; (2) if the -40°F flow requirement failed; (3) if the P. I. failed to exceed the stored P. I. by ϵ. If the count in any of these three places exceeded constants chosen a priori, the run was halted. Tests 1 and 2 were incorporated to halt the program if it reached a situation where it could not pass any of the tests. After the initial program was operating properly, this situation was never encountered.

LISTING FOR NO. 2 COMPUTER PROGRAM

```
C     OPTIMAL DESIGN OF AN ACTUATING SYSTEM                               MAIN0000
      DIMENSION FJS(6),RN(7),OLDD(7),OLDX(7),D(7)
      COMMON  /DATA/ A(38)                                                MAIN0006
      COMMON  /NOMI/ TNOMW ,TNOMC,TNOMV,TNOMR,TNOMDY ,TNOMEN              MAIN0009
      COMMON  /OUP/  TOTW,TOTC,TOTV,TOTR,TOTDYP,TOTENV                    MAIN0012
      COMMON /FLG/ IFPFLG,IERFLG
      COMMON /INTM/ DS1,DS2,DAL1,DAL2, Q, PRES,AREA,R,ACTDP,TUFLW,ANORM MAIN0018
      EQUIVALENCE (A(2),PM),(A(4),PB),(A(5),RM),
     1(A(15),RB),(A(17),AM),(A(23),AB),(A(26),TULAL1),(A(27),TULAL2),
     2(A(28),TULS1),(A(29),TULS2),(A(30),DELMX),(A(31),HM)
      WRITE (6,553)                                                       MAIN0023
      READ (5,500) (A(I),I=1,38)                                          MAIN0024
      WRITE (6,550)  (A(I),I=1,38)                                        MAIN0027
      WRITE (6,555)                                                       MAIN0034
      IFPFLG = 0                                                          MAIN0042
      IERFLG = 0                                                          MAIN0045
      MARK = 15
      PIDI = 0.0
      NO = 0
      NUM = 0
      KOUNT = 0
      PRES = A(1)
      R = A(3)
      DS1=A(8)
      DS2=A(12)
      DAL1=A(6)
      DAL2=A(10)
    2 AREA = HM/(PRES* R)                                                 NOMS0021
      ANORM = AREA                                                        NOMS0121
      ZZ8 = PRES - 3000.
      IF(ZZ8.LT.0.0) ZZ8 =0.0
      ZZ1 =  .001705*( TULS1*(DS1)**(-4.93)*(1.+.0002*ZZ8)**2 + TULAL1*(
     1DAL1)**(-4.93))
      ZZ2 =  .001705*( TULS2*(DS2)**(-4.93)*(1.+.0002*ZZ8)**2 + TULAL2*(
     1DAL2)**(-4.93))
      ZZ5 =  .004533*DELMX                                                NOMS0011
      Q = ZZ5*AREA*R                                                      NOMS0043
      ZZ3 =  (4.4* Q)**1.75                                               NOMS0045
      TUDP  = ZZ1*(8.8*Q)**1.75 + ZZ2*ZZ3                                 NOMS0048
      TUDP2  = ZZ1* ZZ3 + ZZ2 *(2.2*Q)**1.75                              NOMS0051
      ACTDP = PRES - .25*( PRES - TUDP ) - TUDP2                          NOMS0054
      ZZ4 = ( .75*PRES - ACTDP)
      IF (ZZ4.LE.100.0) GO TO 30                                          NOMS0067
      PRES = PRES + ZZ4/ 2.0                                              NOMS0063
      GO TO 2                                                             NOMS0066
    4 DO 16  J=1,7
      SUM12 = 0.0
      DO 15  I=1,12
      CALL RANDM1 (MARK,X)
      RAND = X
      SUM12 = SUM12 + RAND
   15 CONTINUE
      RN(J) = SUM12 -6.0
      D(J) = CO* OLDD(J) + C1 * OLDX(J)
      OLDD(J) = D(J)
      RN(J) = D(J) + RN(J)
      OLDX(J) = RN(J)
   16 CONTINUE
      PRES = PRES + RN(1)* PM
      R = R + RN(2) * RM
      AREA = AREA + RN(3)* AM
```

```
      ARTEST=PRES *AREA*R
      IF(ARTEST.LT.HM) GO TO 220
      PART = 0.0
      DS1=A(8)
      IF(RN(4).LT.PART) DS1=A(9)
      DS2=A(12)
      IF(RN(5).LT.PART) DS2=A(13)
      DAL1=A(6)
      IF(RN(6).LT.PART) DAL1=A(7)
      DAL2=A(10)
      IF(RN(7).LT.PART) DAL2=A(11)
   23 ZZ1 = .3366*( TULS1*(DS1)**(-3.98)*(1.+.0002*ZZ8)**2 + TULAL1*(DAL
     11)**(-3.98))
      ZZ2 = .3366*( TULS2*(DS2)**(-3.98)*(1.+.0002*ZZ8)**2 + TULAL2*(DAL
     12)**(-3.98))
      ZZ5 =  .004533*DELMX                                              NOMS0011
      Q = ZZ5*AREA*R                                                    NOMS0043
   25 TUDP4    =  Q* ( 2.2*ZZ1 + ZZ2)                                   MAIN0136
      IF(PRES.LT.TUDP4) GO TO 222
      ZZ1 =   .001705*( TULS1*(DS1)**(-4.93)*(1.+.0003*ZZ8)**2 + TULAL1*(
     1DAL1)**(-4.93))
      ZZ2 =   .001705*( TULS2*(DS2)**(-4.93)*(1.+.0003*ZZ8)**2 + TULAL2*(
     1DAL2)**(-4.93))
      ZZ3 =   (4.4* Q)**1.75                                            NOMS0045
      TUDP   = ZZ1*(8.8*Q)**1.75 + ZZ2*ZZ3                              NOMS0048
      TUDP2   = ZZ1* ZZ3 + ZZ2 *(2.2*Q)**1.75                           NOMS0051
      ACTOP = PRES - .25*( PRES - TUDP ) - TUDP2                        NOMS0054
      IF((.75*HM-1000.)-ACTOP*AREA*R) 30,30,356
   30 WRITE (6,554)  PRES,AREA, Q, TUDP,TUDP2,ACTOP,R
      WRITE (6,560) (O(I),I=1,7),(RN(J),J=1,7)
      NO = 0
      NUM = 0
      CALL WEIGHT                                                       MAIN0138
      IF( IERFLG .EQ. 0) GO TO 35                                       MAIN0141
   32 PIOX =  0.0                                                       MAIN0144
      IERFLG = 0                                                        MAIN0147
      IF (IFPFLG .EQ. 0) GO TO 210                                      MAIN0148
      GO TO  356
   35 CALL COST                                                         MAIN0150
      IF ( IERFLG .NE. 0) GO TO 32                                      MAIN0152
      CALL DYNPER                                                       MAIN1152
      IF ( IERFLG .NE. 0) GO TO 32                                      MAIN2152
      CALL VOLUME
      IF ( IERFLG .NE. 0) GO TO 32
      CALL  RELIAB
      IF ( IERFLG .NE. 0 ) GO TO 32
      IFPFLG = IFPFLG + 1
  110 PIOX = 0.0                                                        MAIN0153
      FJS(1) = 1.0 - TOTW/(2.0*TNOMW)                                   MAIN0154
      FJS(2) = 1.0 - TOTC/(2.0*TNOMC)                                   MAIN1154
      FJS(3) = 1. - TOTDYP
      FJS(4)= 1.0-TOTV/(2.0*TNOMV)
      FJS(5) = 1.0 - TOTR/ (2.0 * TNOMR)
      DO  112  II = 1,5                                                 MAIN6154
      IF ((FJS(II).LT. 0.0).OR.(FJS(II).GT.1.0)) GO TO 115              MAIN7154
  112 CONTINUE                                                          MAIN8154
      DO  113  II = 1,5                                                 MAIN9154
  113 PIOX =  PIOX +  A(II+31)*FJS(II)                                  MAIN0155
      POT= PIOI + 0.02
      IF(PIOX.GE.POT) GO TO 114
      CO =0.6
```

```
      C1 = -0.7
      KOUNT = KOUNT + 1
      PRES = PRES - RN(1) * PM
      R = R - RN(2) * RM
      AREA = AREA - RN(3) * AM
      IF(KOUNT.EQ. 500) GO TO 210
      GO TO 115
  114 PIDI = PIDX
      CO =0.9
      C1 =0.2
      KOUNT = 0
  115 II=1
      IF(DS1.EQ.0.750) II=2
      IJ=1
      IF(DS2.EQ.0.625) IJ=2
      IK=1
      IF(DAL1.EQ.0.750) IK=2
      IL=1
      IF(DAL2.EQ.0.625) IL=2
      WRITE  (6,552)  IJ,IL,II,IK,    PIDX,TOTW,TOTC,TOTDYP,TOTV,TOTR      MAIN0155
      IF(IFPFLG.NE.1) GO TO 4
      PIDI=0.0
      DO 165 K = 1,7
      OLDD(K) = 0.0
      OLDX(K) = 0.0
  165 CONTINUE
      PRES = 6817.
      AREA = 3.19
      R = 7.7
      GO TO 23
      WRITE (6,551) PIDI
      STOP                                                                 MAIN0195
  220 WRITE (6,556) PRES,AREA,R
      NO =NO+1
      WRITE (6,560) (D(I),I=1,7),(RN(J),J=1,7)
      IF(NO.NE.60) GO TO 356
  221 STOP
  222 II=1
      IF(DS1.EQ.0.750) II=2
      IJ=1
      IF(DS2.EQ.0.625) IJ=2
      IK=1
      IF(DAL1.EQ.0.750) IK=2
      IL=1
      IF(DAL2.EQ.0.625) IL=2
      WRITE (6,557) IJ,IL,II,IK, PRES,AREA,R
      NUM =NUM + 1
      IF(NUM.NE.40) GO TO 356
      GO TO 221
  356 PRES = PRES - RN(1) * PM
      R = R - RN(2) * RM
      AREA = AREA - RN(3) * AM
      CO =0.6
      C1 = -0.7
      GO TO 4
  560 FORMAT ( 7F10.3 )                                                    MAIN0198
    0 FORMAT(1H0,47X,10HINPUT DATA////13H P NOM      = F10.3,8H PM    =    MAIN0204
     1F10.3,10H R NOM   = F10.3,8H PB     = F10.3,3H RM    =               MAIN0207
     2F10.3//13H DAL11       = F10.3,11H DAL12    = F10.3,10H DS11     =   MAIN0210
     3F10.3,10H DS12    = F10.3,11H DAL21     = F10.3,11H DAL22     = F10.3//MAIN0213
     413H DS21       = F10.3,11H DS22     = F10.3,10H W MAX   = F10.3,     MAIN0216
```

```
     58H RB    = F10.3,11H C MAX    = F10.3,SH AM    = F10.3, //13H PUV MAXMAIN0219
     6    = F10.3,11H RED MAX = F10.3,10H R MAX  = F10.3,10H R MIN  =    MAIN0222
     7F10.3,11H DYP MAX = F10.3,  8H AB    = F10.3, //13H ENV MAX   =    MAIN0225
     8F10.3,11H ENV MIN = F10.3,10H TULAL1 = F10.3,10H TULAL2 = F10.3,  MAIN0228
     911H TULS1    = F10.3,11H TULS2    = F10.3//13H DELTA MAX = F10.3,  MAIN0231
     111H HM       = F10.3,10H WTF1    = F10.3,10H COSTF2 = F10.3,11H DYPFMAIN0232
     23    = F10.3//13H VOLF4       = F10.3,11H RELF5    = F10.3,10H ENVF6  MAIN1232
     3= F10.3,11H REL MAX = F10.3)                                       MAIN2232
 551 FORMAT(1H0,47X,28H THE STORED VALUE OF P.I. = F10.3)
 552 FORMAT (1H , 6H CASE 4I1,6X,7HP.I. = F7.2,3X,5HWT = F8.1,3X,7HCOST
     1 = F7.0, 3X,6HDYP = F6.2,3X,6HVOL = F6.2,3X,6HREL = F7.1)
 553 FORMAT (1H1, 40X, 24HOPTIMAL ACTUATING SYSTEM  )                    MAIN0241
 554 FORMAT(1H0, 8H PRES = F10.3, 8H AREA = F10.3, 5H Q = F10.3,         NOKS0075
     19H TUDP  = F10.3,10H TUDP2  = F10.3,9H ACTDP = F10.3,5H R = F10.3)
 555 FORMAT(1H1,34X,25HCASE = DS2,DAL2,DS1,DAL1 )                        MAIN1243
 556 FORMAT (1H0,8H PRES = F10.3, 8H AREA = F10.3, 5H R = F10.3)
 557 FORMAT (1H0,6H CASE 4I1,6X,8H PRES = F10.3, 8H AREA = F10.3,
     15H R = F10.3)
 560 FORMAT (1H ,3H7DS 7F8.2,5X,4H7RNS 7F8.2)
     END                                                                 MAIN0244
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13. ABSTRACT

The servo actuating and corresponding energy distribution subsystem comprise approximately 80% of an aircraft's flight control system weight. Consequently, whenever subsystem tradeoff studies are conducted it is desirable to select the optimum design with respect to weight and other similar parameters. This study investigates and develops such an optimal design process. A sample problem was selected and an optimal technique formulated and applied to the problem. The sample problem was a fixed-configuration hydraulic actuation and power system. The study objectives were to optimize weight, dollar cost, size, dynamic performance, and reliability as a function of the system's independent design parameters. The parameters included pressure, actuator area, actuator torque arm, and plumbing tube sizes. Parameter optimization was accomplished by fixed grid and random searching techniques. Within the framework of parameter optimization, a design philosophy was formulated which allowed dissimilar terms (e.g., weight in pounds and dollar cost in dollars) to be combined to form a total performance criterion for the system. When the optimization technique was applied to the sample problem, the performance criterion showed little variation as a function of the parameters being optimized. All of the cost functions had large nominal values and only slight variations about this nominal. To realize the potential of the design technique developed in this study, different design concepts and possible variations of each one should be considered, to reach a more meaningful optimum design.

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